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Heating Piping & Air Conditioning

DESIGN
INSTALLATION
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MAINTENANCE

JULY ★ 1942

INDUSTRIAL
COMMERCIAL
AND LARGE
BUILDINGS

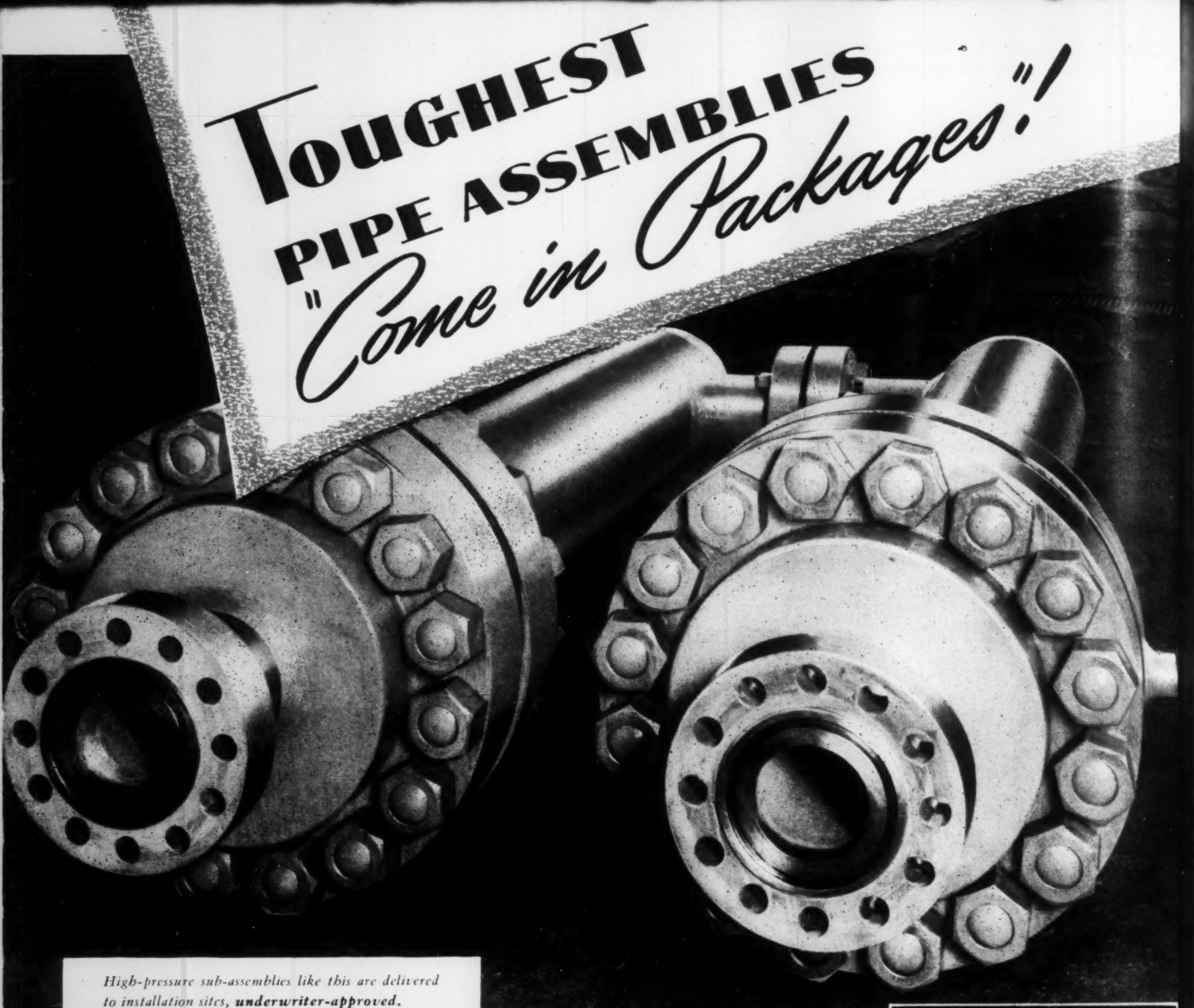
Page 431: Journal Section, American Society of Heating and Ventilating Engineers

JUL 8 1942

"United We Stand"

CONTENTS . . PAGE 43

TOUGHEST PIPE ASSEMBLIES "Come in Packages"!



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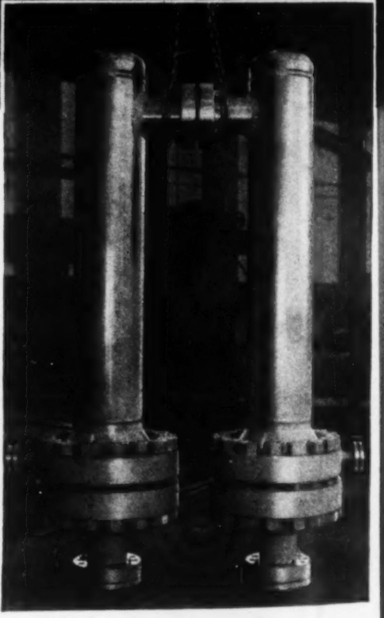

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IN THIS ISSUE

Volume 14 July, 1942 Number 7

The Editor's Pages (information on priorities and other news) . front section 39

Three-Zone Air Conditioning System Serves Shop, Offices, and Drafting Room (flexibility features design of system for war production plant), by T. H. Mabley 401

WPB Permits Assembly of Small Stokers 402

District Heating Association Considers Wartime Problems (summarizing information presented at NDHA convention), by Glen D. Winans 403

War Problems Forum (aircraft engines breathe conditioned air while on test . . . conservation of critical materials in design . . . 13 ways to save oxygen in welding and cutting . . . coal conservation . . . an alloy steel laboratory . . . allocation classifications . . . and others) 405

Fire Protection Association Amends Rules on Air Conditioning and Welding (report of actions taken at meeting of NFPA) 409

Code for Pressure Piping (materials, their specifications, and identification, by F. H. Morehead . . . how a sectional committee functions in formulating a piping code, by E. B. Ricketts) 410

Air in Air Raid Shelters (importance of adequate ventilation), by Walter L. Fleisher 413

Insulation of Cold Pipe Lines (HPAC data sheet) 415

"Open for Discussion" (sizing vent piping for safety valves) 416

How to Protect Equipment Against Failure of Flow (use of orifices and differential pressure switches in chilled water and other piping systems in war production plants), by William Goodman 417

Dormitories for War Workers To Have Forced Warm Air Heating (preliminary design of "duration" dormitories) 422

Flow of Solids in Piping (practical data on pumping and piping sludges, muds, and suspensions in industrial wartime processes), by H. E. Babbitt and D. H. Caldwell 423

How Can I Figure the Heat Transfer Coefficients for These Walls? (Paul D. Close answers a reader's question) 427

Air Filters Cover Entire Ceiling of Production Paint Shop 428

Booster Triples Capacity of 36 Year Old Boiler 429

Air Conditioning Speeds Torpedo Boat Construction 429

Ventilation of Rotor Casting Department Given Special Attention by Motor Maker 430

Equipment Developments (new product information) .... back section 113

Useful Information (recent books and reports) .... back section 131

Trade Literature (helpful new catalogs) .... back section 136

Coming Events (to mark on your calendar) .... back section 142

Index to Advertisers .... back section 144

Journal Section of the American Society of Heating and Ventilating Engineers

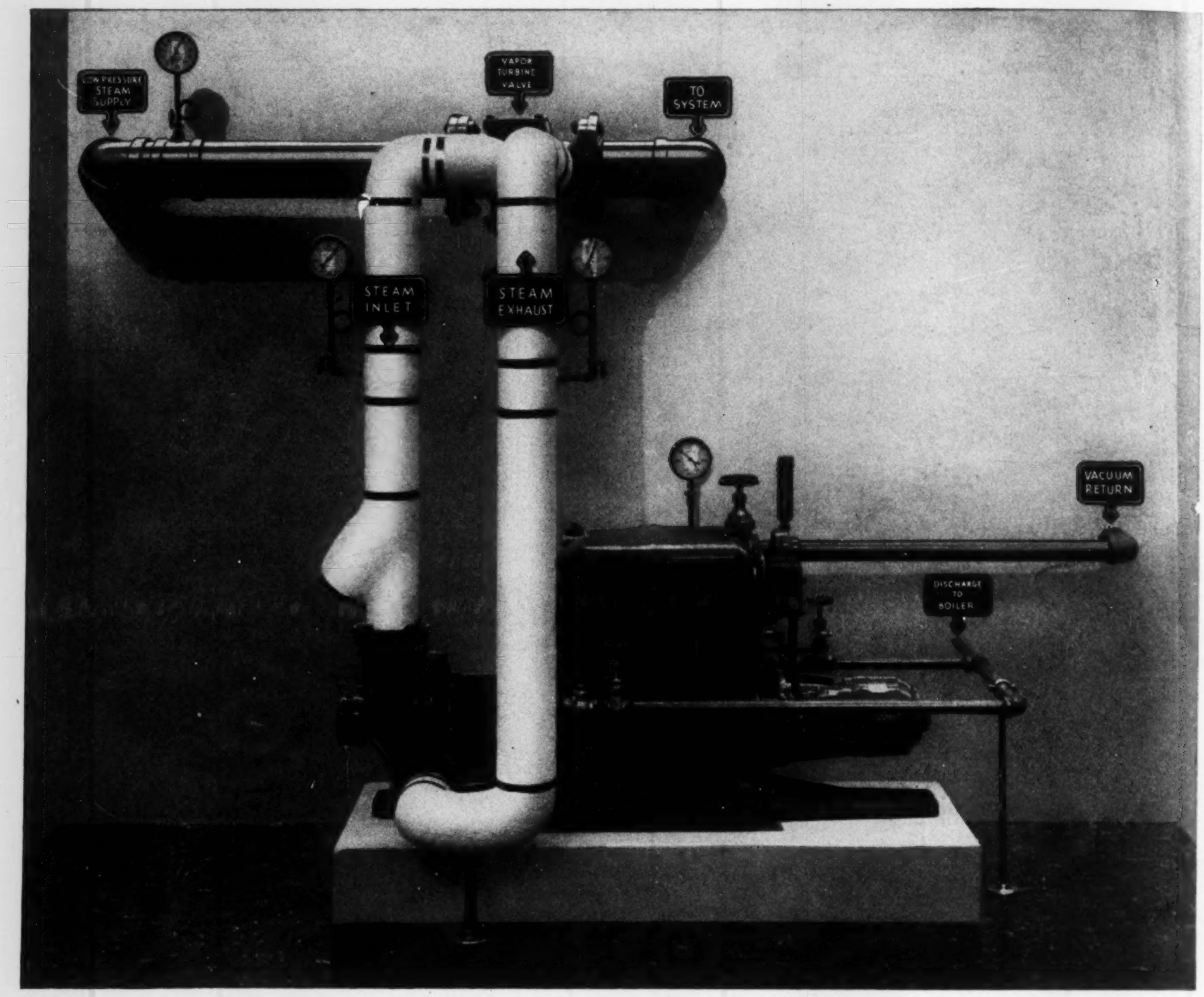
Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson 432

Overloading of Viscous Air Filters During Accelerated Tests, by Frank B. Rowley and Richard C. Jordan 438

Graphical Symbols for Drawings 448

Bulletin Section 453

Interrupted Electric Service wil not stop this NASH Heating Pump



In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is independent of electric current failure, and continues to operate as long as there is steam in the system.

This is because the prime motive power of this economical pump is a special steam turbine, controlled by a unique "Vapor Turbine Valve", which automatically bypasses from the heating main a small portion of steam, the exact amount necessary to develop the power needed to remove the condensate and maintain the required vacuum on the system. Even this small amount is passed immediately back to the mains, and goes on to the system with little heat loss. This pump operates on any system, high or low pressure.

The Vapor Turbine is a most economical pump, for the elimination of electric current does away with current cost, the largest single item in the operation of an ordinary return line heating pump. Bulletin on request.

THE NASH ENGINEERING COMPANY

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Three-Zone Air Conditioning System Serves Shop, Offices, and Drafting Room

T. H. Mabley Describes Installation Designed to Meet Varying Load Conditions at War Production Plant

SUMMARY—Flexibility of operation is provided in the air conditioning system for an industrial building engaged in war production work by separate zoning of the shop and office areas and the large drafting room. The central heating and cooling plant comprises two 15 hp and one 5 hp condensing units, and a direct fired air heater burning No. 5 oil. . . Cost of the complete air conditioning installation was in proportion but little more than for straight heating and ventilation, according to the author, engineer with Mechanical Heat & Cold, Inc.

ONE OF the industrial buildings recently completed in accordance with defense recommendations of military and other authorities incorporates a number of features specifically designed for protection against saboteurs as well as to facilitate blackout. One such feature is the use of small light-ports in the side walls; otherwise the building has no windows. The ports are built of unbreakable glass, are not large enough to invite fire bomb throwing, and give the occupants an opportunity to peek at the weather if they desire. Shutters are furnished for complete blackout.

Three Zones

The building has a complete air conditioning system divided into three independently controlled zones, to provide accurate temperature control irrespective of varying load conditions.

The first zone is the office area, including private and general offices situated along the front of the building. The second zone is the drafting room, and the third is the shop area. Because of the varying loads in each zone, a separate thermostat and exhaust control is preferable for each.

The entire one-story building is served by a single heating and cooling system placed in a corner of the basement. From this central system supply air is furnished to the three zones by means of ducts in the attic space which lead to ceiling diffusers situated neatly between the light troughs in the ceiling. Recirculation of air for the entire system is accomplished by means of a common return trench carried under the floor along the outside wall on three sides of the building. This trench picks up return air from openings distributed along these walls and returns all recirculated air to the central conditioning unit. The conditioning unit includes a single blower, a filter bank, an oil fired direct air heater, and three refrigeration condensing units.

Flexibility

Three condensing units are used for the sake of economy, since this system must operate over a much longer cooling period than does the ordinary commercial air conditioning job. Two 15 hp condensing units serve the drafting room and shop zones. In the summer, both machines are run to provide the necessary cooling effect under maximum load; during milder weather, only one machine is necessary to maintain comfortable conditions. A manual changeover permits the changing of sequence of cycling on these machines so as to equalize their running times. Another advantage of the dual machine installation is reliability, since air conditioning is essential to the maintenance of production and for any use of the building.

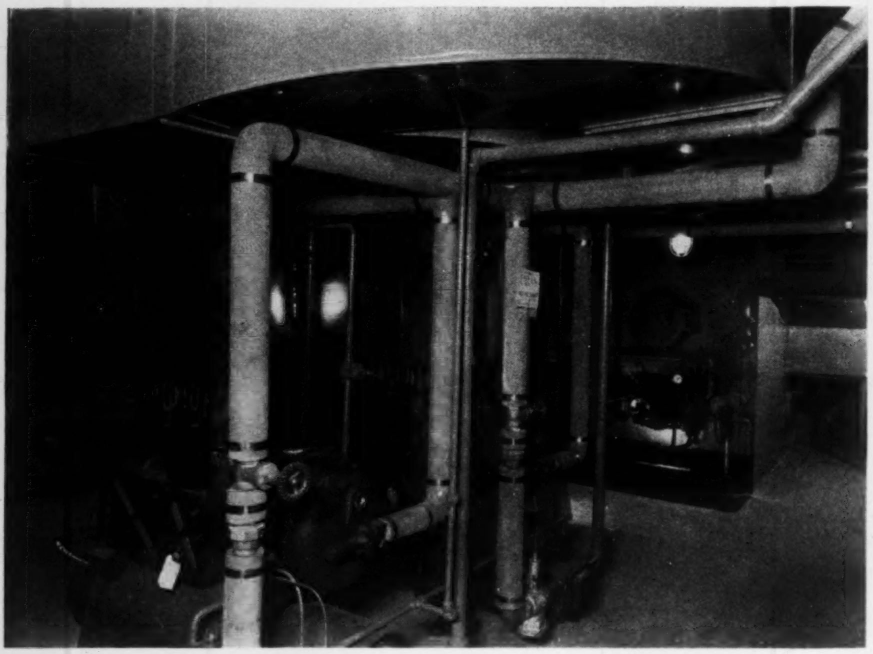
The third 5 hp condensing unit serves the office area. This area is often used for a longer week and many times is occupied when the shop and drafting room are not.

Direct Air Heater

The direct air heater operates with a commercial rotary gun type oil burner using No. 5 fuel oil. The oil storage tank is underground.

The oil is preheated before going to the burner by means of a special heat exchanger built around the breeching from the furnace. The use of this duct air heater simplifies the heating plant and reduces the total space required for the equipment. The heater is of welded steel design with a rectangular combustion chamber in the front of the unit

There are three zones—this large drafting room, the office area, and the shop



The two 15 hp condensing units and the oil fired direct air heater with bypass ducts on each side, are shown in this centralized equipment room. Blowers and filters are back of the heater

which is lined inside with refractory brick. A tubular heat exchanger built in conjunction with the round downdraft flues provides additional secondary heating surface.

Control

The general design of the system is based upon maintaining a constant air circulation at all times and varying the discharge temperature for each zone independently, depending upon the zone heating or cooling load requirements. Constant air circulation is obtained by operation of the central blower. Temperature is controlled by the use of face and bypass dampers around the heating unit. The cooling coils are installed in the bypass connection so that the same temperature regulation system operates automatically throughout the year without the necessity of seasonal change over.

The equipment is controlled from a central panel conveniently situated in the equipment room, which is kept under lock and key. The control system is further protected from tampering by means of locking devices furnished with the instruments. A master clock controls the sequence of the conditioning system operation so as to cut off the cooling system in the summer during the hours that the building is not occupied. At the same time, this clock starts the cooling system early enough in the mornings to assure comfortable conditions at the time the building is to be used. During the winter, the time control also reduces the temperature during the night and weekend shutdown periods to reduce heating costs. Manual control switches are provided to shut off any of these automatic controls in the event of an emergency or overtime working schedule.

A manual control is provided in each zone for the regulation of the exhaust vents. This permits independent regulation of exhaust air depending upon conditions. This is particularly necessary since smoking is allowed in some areas. A damper control is provided for the regulation of the volume of outside air to the central system.

It is interesting to note that air conditioning not only is essential to maintain comfortable working conditions in this building, but that the initial cost for a system of this type was only a little more than that for a good, conventional heating and ventilation system. Operation of the building has proved that the additional investment for year around temperature and humidity control as well as for heating and ventilation has been many times warranted. The system was engineered and designed for the industrial building contractor, the Clausen Co., by Mechanical Heat & Cold, Inc.

WPB PERMITS ASSEMBLY OF SMALL STOKERS

The War Production Board acted June 4 to speed the conversion of oil burning equipment to coal by permitting the assembly of small stokers (class B) from materials which were in manufacturers' hands on May 31. At the same time, W. W. Timmis, chief of the plumbing and heating branch, announced the creation of an operations section which will continue an overall study, underway in the branch for some time, of the various problems involved in the conversion of oil burners to use of other fuels.

Permission to assemble small coal stokers is contained in amendment 1 to limitation order L-75. The original terms of the order ended the production of such stokers on May 31. The amendment, which was approved by the Office of Petroleum Coordinator and the Office of Solid Fuels Coordinator, permits the assembly until September 30 of small stokers composed wholly of fabricated parts in a manufacturer's physical possession on the former cutoff date. The plumbing and heating branch estimated that about 8000 stokers will be assembled under the terms of the amendment.

All stokers have recently been released from the "freeze" provisions of limitation order L-79, making an estimated 70,000 units available for conversion purposes. The possibility of allocating sufficient amounts of pig iron and scrap metal for the manufacture of grates is being studied. A number of foundries are already producing a special pattern of grate designed to fit all types of small furnaces and boilers.

The new operations section of the branch, headed by Henry S. Norris, will have the major responsibility of handling the demand for stokers, grates and other equipment needed for conversion to coal.

The operations section will handle appeals from limitation orders concerning the heating and plumbing industry, and will also be responsible for special assignments, including the continuance of the simplification program.

District Heating Association Considers Wartime Problems

By Glen D. Winans, The Detroit Edison Co., Past President of the National District Heating Association

SUMMARY—The NDHA's 33rd annual convention, held last month, was a streamlined affair devoted principally to discussion of problems brought about by the war. Mr. Winans highlights the various reports and papers here. Information is included on blackout provisions and protection of heating plants in wartime, heat saving methods, equipment changes due to materials scarcities, steel heat exchange surface, operating statistics, etc.

The National District Heating Association held its 33rd annual meeting at Dayton, Ohio, June 11 and 12. The meeting was streamlined to two days and considered, for the most part, the problems brought about by the war.

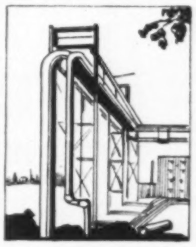
Chester F. Ogden, assistant purchasing agent of The Detroit Edison Co., explained priorities as they apply to the district heating industry. At present the limitation on extensions is $1500 in material and 250 ft in length without permission of the WPB. J. V. Redfield, of Detroit, told of the problems to be expected in obtaining replacement parts or new material for equipment on customers' premises. Mr. Redfield urged "hand to mouth" buying rather than waiting until requirements for material became large.

Blackouts

Blackout provisions and protection of heating plants and systems against the possibility of air raids was given considerable discussion in the report of the steam station engineering committee, by E. E. Dubry for A. S. Griswold.

In Detroit, immediate provisions have been made to blackout all plants and some measures have been taken to provide protection for personnel and equipment against direct hits of incendiary bombs and flying fragments of high explosive bomb near-hits. Blackout of plant buildings has been obtained, in general, by reducing interior illumination. Battery operated emergency lighting systems with low intensity bulbs are the chief source of light. Some use has been made of light weight screens for obscuration where the normal lighting system is in use. Heavy fireproof canvas curtains are also used to provide obscuration and protection against flying glass. Windows adjacent to rotating machines have been heavily planked, and vulnerable skylights have been covered with a 4 in. concrete slab. Fire equipment has been augmented with additional hose, sand, shovels, and extinguishers.

Locations have been designated for the personnel which will provide maximum shelter during an air raid. Each man has been provided with a safety helmet and goggles. First aid classes are being held and medical supplies have been built up. Transportation of casualties has been provided for by rigging a panel truck to carry four Army stretchers.



Some thought has been given to future problems which might arise if the picture changes for the worse. Additional measures could be taken without much delay.

In most cities, rigid enforcement of plant entry rules has been put into effect. Identification badges have been issued to all authorized employees and any visitors have to identify themselves properly before entering. Doors and windows, where possible entry could be made, have been screened or barred. Blackouts have been effected by reducing interior illumination and by obscuration through painting of outside windows with special black paint. Fire fighting equipment has been augmented, especially for fighting incendiary bombs. Employee training and instruction to provide for meeting these new emergencies have been followed generally. Surveys have been made of plant properties to find shelter for personnel during an air attack. Some plans have been made to take care of possible air raid damage.

To protect against air raid damage, the steam distribution system of The Detroit Edison Co. has been zoned into 12 separate zones. Each is fed by a separate feeder pipe from the plant. In case of damage, this zoning permits shutting down smaller portions of the system at the plant than normally would be possible.

Customers have been instructed to shut off steam in the case of an actual air raid or in case of a fire. In case the customer wishes to restore service, he is instructed to see that temperature controls and vacuum pumps are operating. Service valves and control valves have been tagged for the customers' convenience.

Most companies have organized emergency crews with the necessary equipment to handle air raid damage.

Saving Heat

The report of the commercial relations committee was given by G. H. Tuttle, of Detroit. This report consisted of thumbnail sketches on heat conservation as follows:

1) Description of the booklet, Principles of Economical Heating, recently published by the National Association of Building Owners and Managers in cooperation with the NDHA.

2) Methods of sizing economizers, by A. F. Metzger, Allegheny County Steam Heating Co. In sizing economizers, the various methods in common use produce widely different requirements for heating surface. It was pointed out that different types of buildings would have different requirements as to heating surface and storage capacity. More research work is required on this subject.

3) Open jet steam for humidification, by H. L. Warhanek, Wisconsin Electric Power Co. He pointed out the difficulty of measuring steam used by radiator air vent humidifying devices.

4) Steam requirements for process steam, by R. M. McQuitty, Union Electric Co. of Missouri. The data on steam

requirements for process equipment consisted of an extensive compilation of information from various sources, with figures on operating pressure; pounds of steam use per hour, per week, and per month; and maximum steam consumption, for cleaning and pressing equipment, laundry equipment, kitchen equipment, hospital equipment, and various industrial processes.

The shortage of oil and its effect on the companies on the East Coast was discussed in the report of the sales development committee given by Leonard S. Phillips of the New York Steam Corp., retiring national president of the NDHA.

J. Earl Seiter, Consolidated Gas Electric Light and Power Co. of Baltimore, chairman of the rates and regulations committee, explained that many companies were considering rate increases due to increases in operating cost. These for the most part are in the form of coal clauses. In some cases there will be tax and labor clauses.

The chemistry committee report was presented by Leo F. Collins, Detroit, its chairman.

Equipment Changes

Equipment, design, and construction changes brought about by scarcity of materials was discussed by Mr. Tuttle. He pointed out that all unit heaters, convector coils, and water heaters are 100 per cent frozen for they can be furnished only for high priority order projects on specific release by WPB. He said that specifications for unit heaters by the procurement branches of the War Department, Navy Department, and other government agencies prohibit the use of copper and other nonferrous materials other than in motors and controls, except:

The Navy bureau of ships; the Maritime Commission; the Coast Guard; and the marine division of the Army, will continue to specify units having all copper or brass heat exchange elements for *shipboard* uses.

The Navy bureau of yards and docks will specify copper tube, steel fin units for installation outside (only) of the continental United States. Alaska, in this case, is considered to be *outside* territory.

Manufacturers have changed to steel tubes, fins, and headers except those sold under the above specifications. There has been no change in basic design except diameter and thickness of tubes, and thickness and spacing of fins.

Manufacturers of heat exchange surface equipment recognize the fact that steel is not as efficient a heat conductor as copper and have therefore increased the radiating surface for a given size unit. Tube thicknesses have been increased to offer more resistance to corrosion failures.

Most manufacturers admit that steel tubes will be more susceptible to oxygen and carbon dioxide corrosion than copper; however, they maintain that steel will outlast the emergency and in that case the units will have accomplished their purpose. Others insist that inasmuch as the life of the coil is a function of the corrosive properties of the steam and condensate, and if the corrosive action is high, even copper coils would eventually fail. All seem to agree that *so-called* adequate venting and sufficiently large flow areas to insure free drainage will reduce internal corrosion.

Manufacturers of water heaters anticipate the life of steel elements to be approximately 50 per cent of that of copper elements. In some cases, the tube thickness has been increased to twice that of the former copper tube.

Dr. E. W. Guernsey, Consolidated Gas Electric Light and Power Co., of Baltimore, discussed the substitutes which could be used for boiler water treating chemicals should there be a shortage in the supply of those used at present. Mr. Collins gave a brief talk on the corrosion studies being made by the various agencies, such as the ASHVE and others.

The report of the research committee was made by G. K. Saurwein, Harvard University. A paper on the uses of cell concrete was given by H. W. Frederick, Cornell University. He gave a resumé of the literature on cell concrete and suggested its use in case of shortages of other insulating materials. The discussion indicated there is no immediate possibility of insulating material shortage.

S. S. Sanford, Detroit, gave a description of what was being done to standardize weather bureau data. This work is being done in cooperation with the American Gas Association, American Society of Heating and Ventilating Engineers, National Warm Air Heating and Air Conditioning Association, U. S. Weather Bureau, Oil Burner Institute, and the NDHA.

Considerable discussion was occasioned by the report on building volume measurements by W. F. Friend and R. E. Hansen, of Ebasco Services, Inc. The consensus was that the building volume measurements should have more study, as there seems to be considerable disagreement about the most accurate unit of measurement to be used in comparing steam consumptions.

Operating Statistics

T. B. Brown, Jr., New York Steam Corp., presented the operating statistics committee report. Major developments of the year 1941 were, according to the analysis:

1) A general decrease in sales volume for almost all companies, with the exception of several companies in the Eastern, Mountain, and Pacific districts.

2) A general decrease in revenue tempered somewhat by increased average revenue and increased connected load.

3) An increase in connected load.

4) Much warmer weather for the Eastern and Central districts, which districts represent a greater part of the industry.

5) Much greater increases in cost of fuel per million Btu for most of the companies than has been experienced in many years.

6) A negligible increase in number of employees, but a considerable increase in hourly rate.

7) A slight increase in invested capital.

Officers elected for the coming year are: W. A. Herr, Philadelphia Electric Co., president; J. J. Schenk, Rochester Gas & Electric Co., first vice-president; J. M. Arthur, Jr., Kansas City Power & Light Co., second vice-president; and S. S. Sanford, The Detroit Edison Co., third vice-president. Members of the executive committee are J. F. Malone, Consolidated Gas Electric Light and Power Co. of Baltimore; R. M. McQuitty, Union Electric Co. of Missouri; and Leonard S. Phillips, New York Steam Corp. John F. Collins, Jr., is secretary-treasurer.

JOIN THE NAVY
AND
FREE THE WORLD

AIRCRAFT ENGINES BREATHE CONDITIONED AIR WHILE ON TEST

When an aircraft engine is delivered to a plane manufacturer it must be ready to go to work at once. There is no such thing as flying a plane slowly for so many miles or hours until the engine is run in. The first thing that a new aircraft engine is required to do is to operate at maximum power in order to get the plane off the ground and into the air. After this has been accomplished the engine may then be throttled back to cruising speed and perhaps treated with more than usual care until the plane has been landed and is ready for another take-off. The job that must be done in the test department of an aircraft engine plant is to take the engines as they are delivered from the manufacturing department, run them in thoroughly and check all phases of their performance.

Of the various types of test equipment, the feed back system is the only one which offers any sort of a return for the money spent for gasoline. With this system the engine drives, through flexible shaft couplings, a magnetic coupling of the eddy current type. This unit permits the engine to run at whatever speed desired, maintaining all the while a constant speed on the output shaft of the coupling. The next unit is an air operated brake which is mounted on the output shaft of the coupling. Next we have a large alternator of sufficient capacity to absorb all the power the engine is capable of developing and finally at the end the field exciter for the alternator. The electric power generated is fed back to the power supply system.

The engines are brought into the preparation room from the assembly department on monorail units and are installed on the stands on which they are supported while under tests. All the fittings and connections are installed in this room and after the engine is completely "dressed up" the engine and stand assembly are picked up by the monorail unit and carried into the test cell. The assembly is then dropped down in place on the bedplate, the shaft coupling bolted up, and the connections to the test stand instrument, oil and fuel lines rapidly made by means of snap-on couplings of various types.

Below the preparation room is a corridor, extending the full width of the department, which is air conditioned and in which is a series of blowers, each of which supplies air through a venturi air flow meter to the carburetor of an engine. By conditioning the air in this carburetor air chamber we are able to reduce the corrections necessary in calculating our engine performance, as the air is always maintained at a uniform condition.

The test cells themselves are square rooms with heavy concrete walls and with double refrigerator type access doors. A noise seal is used where the coupling shaft projects through the wall of a test cell. As a result of this design the noise level in the department is very satisfactory.

Immediately above the test cells is a penthouse containing the supply fan for cooling air and an exhaust fan which scavenges the test cell. Inlets and outlets of these fan rooms are equipped with generous soundproofing and as a result the noise level outside the building is very satisfactory.

The corridor immediately under the test cells contains the multitude of pipes necessary for supplying oil and water and for connecting the many instruments and gages in the control rooms to the engines on test.—from a paper by C. A. Chayne presented at the Midwest Power Conference.

CONSERVATION OF CRITICAL MATERIALS IN DESIGN

Stephen F. Voorhees, special advisor, construction branch, production division, WPB, and past-president of the American Institute of Architects, speaking of the responsibility of the designer at a meeting held under the auspices of the Producers' Council Club of Washington, D. C., said: "We must eliminate nonessential uses in order to provide now for essential war purposes. If we don't do it now we are not going to reach and serve the ultimate means. We can do it. I don't think designers are in a strange field—the conservation or diversion of critical materials is the designer's problem.

WAR PROBLEMS FORUM

"The time to start to make real conservation is the time you start to design. That is when you give the greatest effect to your effort.

"There are really two methods or times for these savings, though the first will produce the larger results. The first is previous to crystallization of the overall, comprehensive scheme and the second will include those elements which could be left to later determination, even to a change at almost the last minute.

"You can't judge the maximum saving merely by examining the list of materials that are to go in and endeavoring to substitute less essential for more essential materials. The only way to make a success of conservation is at the time and place of the basic design formulation.

"Typical examples in the first class are these:

"A building was designed, and perhaps properly so, to be air conditioned. The windows are much less than is usual, and the spaces from the windows to the deepest point of desk use are excessive. With air conditioning and proper lighting, the entire space will undoubtedly be very habitable. It is too late at this time to change or to leave out the air conditioning unless the owner is willing to occupy only about one-third of the net usable area of the building.

"Another example is a war plant. One of our guests tonight was discussing with a designer of a bomber plant the type of construction of an office building which was tentatively planned to go alongside the great assembly building. The initial conception was that it be of fireproof construction and possibly air conditioned. Our guest didn't know whether air conditioning was essential and wondered if something couldn't be saved by making this office building of something less than fireproof construction, but if so it couldn't be placed alongside this light steel assembly building. The final decision was to move it away from the assembly building and construct it like the temporary buildings here in Washington.

"The second method, or things which can be left for later determination, are such things as cast iron basin cocks, builtup asphalt flashing for copper, wood windows for steel windows, concrete pipe for steel pipe and drainage work, and so on. You are all familiar with many more. Specifications alone won't do the trick. Neither can laws, edicts, or priorities or anything else do the job. But the job will be done if we can get the will and the skill of the designers and the owners and producers of America to put their shoulders to the wheel. It has got to be done now, not a year from now."—from the bulletin of the Producers' Council.

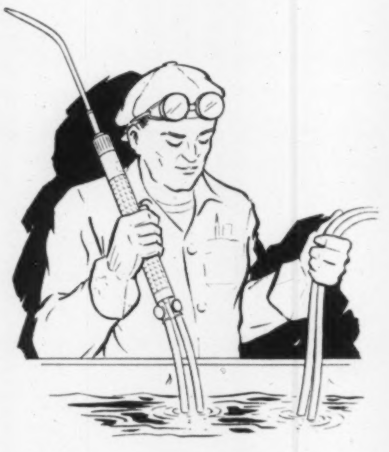
CHROMIUM AND NICKEL FOR INDUSTRIAL PROCESS REGULATORS

Part 1244—instruments, valves, and regulators used in industrial processes—of the WPB's conservation order L-134 restricts the use of chromium and nickel in industrial instruments, instrument ends, instrument connections, control valves, safety valves, and regulators. The order, issued May 26, is completely and specifically phrased, and the allowable uses listed give consideration to chemical reactions, corrosion, and contamination, as well as pressure, temperature, and velocity conditions.

13 WAYS TO SAVE OXYGEN IN WELDING AND CUTTING

1) *Use the proper size torch tip.* A No. 2 tip used instead of a No. 1 for cutting ½ in. plate consumes 10 to 20 per cent more oxygen, 16 per cent more acetylene—all wasted.

2) *Do not use excessive pressures.* This, one of the most common malpractices, is usually done by inexperienced operators who believe they thereby get a more efficient flame, or to save a trip to the pressure chart. Manufacturers' rated pressures are the most efficient for average conditions. And when more speed is desired, the new high speed machine cutting tips cut faster without consuming additional oxygen.



Check hose periodically for leaks. Dip hose in water while under pressure. Use soap water on connections

3) *Close cylinder valves when torch is not in use.* If only the torch valves are closed, gas pressure in the hose rises and may force leaks. If leaks are already present or if new leaks are forced, considerable gas will be lost.

4) *Check hose and connections for leaks.* They may be infinitesimal, but a $\frac{1}{64}$ in. diameter leak in hose operating at 100 psi will waste 290 cu ft of oxygen in 24 hr—more than a tankful. The hose should be checked periodically by immersing it in water while under pressure, to show up leaks. Soap and water will reveal leaks at connections.

5) *Use correct diameter hose.* Hose that is too small for the job requires higher gage pressure to secure sufficient flow, causing unnecessary strain and promoting leakage in the hose.

6) *Use short lengths of hose.* Unnecessarily long hose also requires higher pressure, to overcome pressure drop in the hose. Also, the surplus lengths, though useless for the job, are subjected to unnecessary wear and tear.

7) *Use all the gas in each cylinder before connecting a new one.* Use every cylinder for all it is worth before disconnecting it. In the larger shops interruptions of work to change tanks can be eliminated by manifolding gas cylinders or by installing pipe lines.

8) *Keep cylinder inventories low.* The old custom of keeping plenty of reserve cylinders must be abandoned, for it "freezes" them at a time when rapid turnover is imperative. Gas suppliers are cooperating by providing frequent deliveries. You can help by ordering fewer cylinders oftener, and by keeping reserves of cylinders at a minimum.

9) *Return empty cylinders promptly.* Idle cylinders are production slackers.

10) *Don't leave torches burning when not in use.* It may seem troublesome to be constantly re-lighting the torch, but if it burns idly for even as little as 3 min out of each hour, it is wasting 5 per cent of its gas consumption. Five per cent of last year's industrial oxygen consumption was 360 million cu ft. Let's use those tankfuls for welding and cutting, not for heating the air.

11) *Keep tips clean and free from carbon and slag.* An unclean tip is an inefficient tip, and, like any dull tool, is abhorrent to the skilled workman. Congested orifices nullify the value of using proper pressures and speeds, resulting in lower efficiency and consequent waste of gas.

12) *Don't abuse cylinders.* A damaged cylinder must be taken out of circulation. It contributes nothing while undergoing repairs.

13) *Hose:—Handle with care.* Because hose is made of rubber, it is most important that it be made to last as long as possible. To ensure this, it must be protected against mistreatment which promotes leaks. Therefore, keep hose out of range of hot oxide and sparks; wash off oil and grease, which ruin rubber; place a plank on each side of the hose where it crosses truck runways; store any hose stocks in a cool, dark place; and when a hose is damaged, cut out the bad portion and splice the ends together with standard splicing nipples. —G. V. Slottman, manager, applied engineering dept., Air Reduction Sales Co.

EVERY COAL CONSUMER CAN AID VICTORY BY ALL-OUT CONSERVATION

"EVERY OUNCE of strength this country has should be dedicated to the job of winning this war as quickly as possible. Labor wasted in producing coal that is lost through improper burning—and for that fact, the preventable loss of heat or energy due to other reasons—is labor that might have been devoted to making ships, planes, tanks and guns to end the war quicker. The railroad facilities used to haul the coal that escaped as smoke, might have been used to speed the production of arms, or the movement of arms and troops to the battle fronts. Every minute that victory is delayed means that more boys like yours and mine will die. Every minute by which victory can be speeded means that some boy, who might not have returned otherwise, will come back.

"Take this message, and carry it back to the people of your communities. The little things like saving a few tons of coal may seem insignificant. But they count, now.

"There are no statistics on the total amount of labor and transportation that still is lost through wasteful burning of coal, despite conservation accomplishments to date, but without doubt the amount is high. And, certainly it would be a contribution well worth making in speeding the winning of the war if every person who could would conserve coal. That goes for the householder, who will use coal again next winter, and for the businessman or industrialist who is using it now.

"I do not wish to indict all smoke makers, since some of it is unavoidable. Sometimes it is necessary for factories to use their equipment at a rate exceeding normal capacity, and under conditions where conservation is more difficult. I think that industry in general is conservation conscious. Factories want to save coal because that saves money. But there no doubt is still a great deal that might be accomplished on the industrial front.

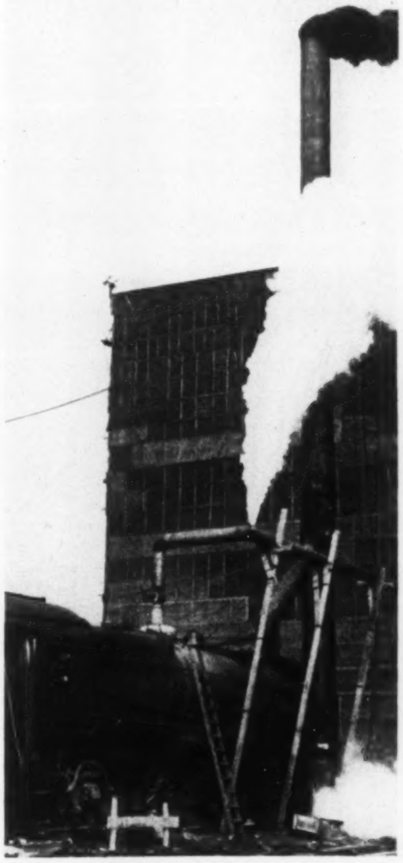
"Great strides have been made in the improvement of coal utilization since the first world war. . . . The manpower and facilities presently available could not supply the coal to meet 1942 energy demands if the fuel still was being consumed at the wasteful rate we used it to comparable results in 1920. The men and materials that have been released to other war purposes by the progress we have made in conservation are a substantial contribution toward winning the war.

"Although coal conservation already is paying big dividends, it is the responsibility of every consumer to make the biggest contribution to the war that he can by saving coal wherever he can.

"We have not reached the point where we must save coal by doing with less heat; and with foresight, we do not expect to reach that point. But, we have reached the point where we must save coal by not wasting it—to conserve manpower, materials and transportation to win the war, quickly and thoroughly."—THOMAS J. THOMAS, associate director in charge of bituminous coal, Office of Solid Fuels Coordination, at the annual convention of the Smoke Prevention Association of America, June 2.

KEEP 'EM HEATING

A retired railroad locomotive, rented for the occasion, supplied steam for heating a new Westinghouse plant last winter. Because speed is the keynote to victory, production at the plant was started before the regular heating system was installed. A special 60 ft stack provided plenty of draft and carried smoke above the building. A regular operating crew of fireman and engineer kept the old locomotive "running."



AIR CONDITIONING BOOSTS OPTICAL WAR GOODS PRODUCTION

THE OPTICAL instruments industry—now being called upon by the government for increasing quantities of lenses, prisms, mirrors, and similar equipment for the armed forces—is finding air conditioning a valuable tool for increasing the speed and quality of production, according to Logan Lewis, vice-president of Carrier Corp. Mr. Lewis lists four major categories in which air conditioning is serving this industry:

1) To prevent uncontrolled expansion and contraction of material.

2) To provide cleanliness, the requirements for which are frequently extreme and always above average.

3) To minimize perspiration which, if transferred to the finely finished metal mounting, may cause an imperfection.

4) To help standardize processes in order to qualify more individual workers drawn from the same labor radius.

One leading optical company controls temperatures in its engraving room within limits of plus or minus ½ F. Here, engraving exceedingly fine lines on glass (sometimes 200 to the inch) calls for elimination of expansion or contraction of both glass and machine. The same company utilizes air conditioning in rooms where grinding and polishing are done and where, in extreme cases, tolerances must be held down to five-millionths of an inch.

Cleanliness and freedom from perspiration required to cut down rejects in cementing and assembly rooms are also provided by air conditioning.

CONTROL OF AIR FLOW VITAL IN ALLOY STEEL LAB

Air and temperature control in a new chemical laboratory building at the Vanadium Corp. of America is facilitated by sealed walls of glass brick and glazed tile, acoustical panel ceilings, and insulating air spaces in the walls and roof. The laboratory was designed and built under the supervision of O. M. Svensson, chief engineer for Vanadium, by Rust Engineering Co.

A positive pressure is maintained within the building, 8 per cent more air being delivered by the heating system into the rooms than is drawn off by the hoods. This produces a flow of air outward from the analytical rooms, through doorways, stair shafts and the storeroom and basement windows, and prevents entry of dust or gases from without.

Of the large amount of air which the hoods must handle to evacuate gases, 75 per cent is supplied by forced air introduced around the upper periphery of the hoods. This air is tempered to 60 F in cold weather. Fuel is saved by this method, because if all this air were furnished by the room heating system it would have to be at a temperature of 70 or 75 F. Only 25 per cent of the air is drawn from the rooms and a draft across the backs and necks of workers is thus avoided. To maintain an excess of air flowing away from the analytical rooms, air equivalent to 33 per cent of that drawn off by the hoods is delivered through the heating system.

The hood design incorporates a double roof and double side aprons, within which adjustable baffles control both the discharge and the characteristics of the air curtain around the sides, keeping the hoods cool. This is particularly advantageous in hot weather. Exhaust piping is held to a minimum to reduce maintenance due to corrosive gases and resistance in removing heavy gases.

The emphasis on atmospheric control arises from the sensitivity to contamination during difficult alloy analyses, such as the determination of gases in steels.

The acid resisting, cast iron pipe, chemical drains are mounted exposed under the basement ceiling where they are easily accessible for repairs. These drains join the sanitary sewers in a manhole outside the building, eliminating numerous troublesome traps.

NEW ALLOCATION CLASSIFICATIONS SIMPLIFY PRIORITIES PROCEDURE

A simple, uniform classification system which will enable the War Production Board to trace the flow of materials in terms of end use, from primary producers to finished products, was announced last month by J. S. Knowlson, director of industry operations. The new allocation classifications, to be used on purchase orders and reports to WPB, will supersede a variety of use classifications now required under materials ("M") orders.

Allocation classification symbols must be used on all orders placed by manufacturers, fabricators, primary producers, etc., after June 30, 1942, and on all such orders which call for deliveries after July 31, regardless of when the orders are placed. To facilitate use of the symbols, they should be placed on purchase orders at an earlier date wherever possible.

Purchases by retailers, and purchases by distributors for resale are exempt from the requirements, which is embodied in priorities regulation No. 10.

Applications under the production requirements plan for the fourth quarter of this year must identify the end use of materials for which application is made by means of this standard allocation classification. The symbols will thus become a principal method of checking the requirements and distribution of materials for war and essential civilian purposes.

The classification itself is comparatively simple. Numbers from 1.00 to 23.00 have been assigned to all major classes of military, industrial, and civilian uses. These broad end use classifications are subdivided as necessary by the use of numbers after the decimal point. For example, class 9.00—power, light, and heat—has under it subclass 9.10, electricity; 9.20, petroleum; 9.30, coal and coke; 9.40, gas. The order of the numbers bears no relation to the relative importance of the use.

In addition, letters known as purchasers' symbols are assigned to broad classes of purchasers, including Army, Navy, foreign purchasers, etc. Both the allocation and purchasers' symbol will be placed on each contract or purchase order, so that the supplier and the War Production Board will be able to ascertain the end use of the material ordered and the classification of the purchaser.

The information obtained by use of the symbols will make possible a more accurate and effective allocation of materials to war requirements and to maintenance of essential civilian industries such as transportation, electric power, health supplies, communications, etc.

All companies required to use the classification should place the appropriate symbols directly on all of their purchase orders if they know the classification for which the material they order is to be used. Intermediate users and suppliers who do not have this information should receive it, as the classification comes into general use, on the purchase orders addressed to them, and should transmit the symbols on their own orders.

EMERGENCY REPAIRS FOR SERVICE INSTITUTIONS

Operators of motion picture theaters, hotels, restaurants and other service institutions should communicate with War Production Board field offices in their localities whenever emergency repair or maintenance materials are needed. "Emergency" means an actual breakdown of existing equipment or a situation where it is evident that a breakdown or suspension of operations is imminent because of damage, wear and tear, destruction, failure of parts or a similar situation.

* * *

According to Associated Press reports last month, which were based on information from German broadcasts, the nazi marshal, Rommel, credited much of his success in Libya to the use of "refrigerated" tanks, with cool interiors despite high outdoor temperatures.

FIRE PROTECTION ASSOCIATION AMENDS RULES ON AIR CONDITIONING AND WELDING

SUMMARY—At its recent annual meeting, the National Fire Protection Association, whose address is 60 Batterymarch St., Boston, Mass., adopted amendments to its standards for the installation of air conditioning equipment, revised standards governing the installation and operation of gas systems for welding and cutting, and revised regulations for foam and carbon dioxide extinguishing systems. . . . Principal changes are presented here in brief form.

Amendments to standards for the installation of air conditioning equipment were adopted by the National Fire Protection Association at its annual meeting held recently in Atlantic City, upon presentation of a report by the committee on air conditioning prepared under the direction of Chairman F. H. Faust of the General Electric Co.

Specifications for the thickness of metal for air ducts were enlarged to include approved types of joint connections and bracing.

A measure of safety in the rating of fusible links for automatic fire doors and dampers was adopted by setting the rating at 50 per cent higher than the maximum temperatures to be encountered in any air conditioning system.

An exception was granted to the rule requiring smoke dampers in the main supply and main return ducts of systems of more than 15,000 cfm capacity when the system is located on the same floor it serves.

Formerly, all electric lamps enclosed in an air conditioning system were required to be of the marine or vaportight type, but a new amendment makes it possible to use germicidal lamps having no filament and of low exposed surface temperatures.

The use of No. 18 U. S. ga metal for louvered type dampers was specified, providing each louver is not more than 6 in. wide and is stiffened by formed edges.

Other amendments concern temperature protection devices for enclosed motors, the adjustment of photoelectric smoke detectors in air ducts, and dampers for supply ducts of high temperature heating systems. In several instances, the wording of sections of the standards was changed slightly in the interest of precision.

Cutting and Welding

Revised standards governing the installation and operation of gas systems for welding and cutting were also adopted by the NFPA. In view of the number of serious fires that have occurred during wartime as the result of careless cutting and welding methods, the revised standard is one of the most important measures enacted by the association during its annual meeting.

The task of revising the standards was undertaken upon request of the International Acetylene Association. They apply to all gas welding and cutting systems and cover all gases used with oxygen for welding, cutting, heating and heat treating operations. The standards do not apply, however, to systems in or upon cars or other rolling stock, or upon vessels.

The installation and operation of stationary acetylene generators are regulated with regard to location, generator rooms or compartments, ventilation, lighting and heating of generator rooms or buildings, and in other important details. Portable acetylene generators, service piping systems for all gases, the manifolding of cylinders to headers for shop pipeline supply systems, storage of cylinders of gases and of calcium carbide, and operating procedure, all are carefully considered and regulated.

The committee on gases of the NFPA also announced that the American Gas Association has requested that revision of the recommended good practice requirements for the installation, maintenance and use of piping and fittings for city gas be undertaken.

Foam and CO_2 Systems

A revision of the standards for foam and carbon dioxide extinguishing systems was adopted by the association upon presentation of a report from its committee on special extinguishing systems, the chairman of which is A. L. Cobb of the Eastman Kodak Co. Such systems are installed in industrial plants to protect special hazards, such as dip tanks and spray booths, and for outdoor hazards such as oil tanks and containers of other flammable liquids.

The foam used in such systems is of two types. One is suitable for use on fires in flammable liquids not miscible with water, such as petroleum and its products. The other is used on fires in liquids that are miscible with water, such as alcohols, esters, and acetone and it may also be used on petroleum products. The foam is supplied from manually controlled pumps, automatic pumps, air pressure tanks, chemically generated pressure tanks, gravity tanks and foam generators. When applied to fire, the foam blankets and smothers it. The standards adopted regulate the details of supply, mixing, discharge outlets, automatic release devices and maintenance.

Carbon dioxide systems are used in many industrial locations and the details of their installation and operation are closely regulated by the new standards. Such systems release carbon dioxide gas which smothers fire by excluding oxygen from it. They operate by completely flooding enclosed spaces with gas, or by local application of gas on particular hazards.

CHARLES E. WILSON ELECTED ACRMA PRESIDENT

Charles E. Wilson, Worthington Pump and Machinery Corp., was elected president of the Air Conditioning & Refrigerating Machinery Association, Inc., at its annual meeting held recently. Other officers are J. P. Rainbault, General Electric Co., first vice-president; W. H. Aubrey, Frick Co., Inc., second vice-president; and P. A. McKittrick, Parks-Cramer Co., treasurer.

S. E. Lauer, York Ice Machinery Corp., was elected chairman of the ACRMA board of directors. William B. Henderson continues as executive vice-president.

CODE FOR PRESSURE PIPING

SECTION 7 —MATERIALS, THEIR SPECIFICATIONS AND IDENTIFICATION

By F. H. Morehead, Chairman, Subcommittee on Piping Materials and Identification; Vice-President, Walworth Co.

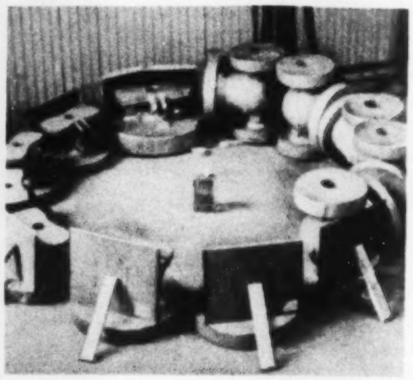
SUMMARY—Since the American Standards Association code for pressure piping (B31) was first issued as a tentative standard in 1935, there have been many developments and advances in piping practices, as described by Sabin Crocker in the January HPAC in the first of a group of articles, of which this is the seventh and last. The piping code has been thoroughly revised and is being reissued as a full American standard. Section 1 of the new code, on power piping, was reviewed by Alfred Iddles in the February HPAC. In March, the section on gas and air piping was covered by John S. Haug, and the section on oil piping systems was analyzed by A. D. Sanderson. The code sections on district heating piping, and on refrigeration piping (including air conditioning) were reviewed by G. K. Saurwein and A. B. Stickney in April. In May, part of the section on fabrication details was discussed by Ludwig Skog, who completed his review last month. . . . This month, F. H. Morehead covers the section on materials, their specifications and identification, and E. B. Ricketts tells how a sectional committee functions in formulating a code of this kind

THIS SECTION of the piping code has been completely revised since publication of the 1935 edition. It serves as an index to all of the dimensional standards to which other sections of the code make reference, as well as to materials specifications and identification.

Included in section 7 are the official designations of all applicable dimensional standards published by the American Standards Association, American Petroleum Institute, the Federal Specifications Executive Committee, and the Manufacturers Standardization Society of the Valve and Fittings Industry. A considerable number of new standards have been published by these various bodies since the previous edition of the piping code, and practically all of the standards have been revised and extended to include information not previously published.

The list of materials specifications has been extended to include materials used in the construction of the piping itself, and also to include many specifications for materials for miscellaneous use in structures other than piping, as for instance, hangers, etc.

Specific requirements are given in section 7 of the code for pressure piping as to marking of various component parts of the piping system. The names and addresses of the standardization bodies whose specifications are quoted in the various sections of the code have also been included in section 7 for convenience in securing copies of the complete specifications.



A "Sunday School" setup during the radiography of steel castings.

HOW SECTIONAL COMMITTEE FUNCTIONS IN FORMULATING A PIPING CODE

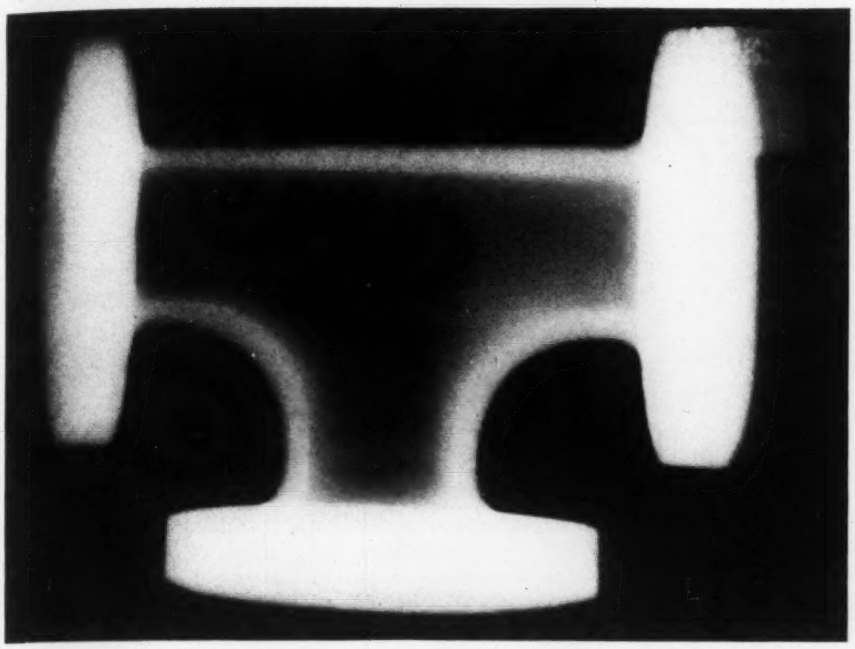
By E. B. Ricketts, Chairman, ASA Sectional Committee on a Code for Pressure Piping; Mechanical Engineer, The Consolidated Edison Co. of New York

A COMMITTEE has been defined as a body that keeps minutes and wastes hours. Since the revision of the code for pressure piping was initiated back in 1937, it might be concluded logically that the sectional committee had indeed kept many minutes and wasted untold hours.

However, over 100 representatives of industry, trade associations, government bureaus, safety councils, etc., make up the sectional committee on a code for pressure piping. The diverse interests and divergent viewpoints on the subject of safety in pressure piping could not be immediately reconciled unless dictatorial powers were granted to some one person. The committee procedure is necessarily slow, since full debate is permitted before a decision can be reached, and frequently a compromise solution results.

Knotty Problems

There are a few prerogatives which the chair can exercise, however, to expedite the business at hand. For instance, when a particularly knotty problem is introduced on which it seems no agreement can be reached, it is common practice in committee procedure to appoint a subgroup to study the matter and submit a report at a future meeting.



Gammagraph demonstrating the soundness of a small standard flanged tee

naming as the chairman of the subgroup the man who initiated the problem. This often has a salutary effect on recalcitrant committeemen, and incidentally, usually produces results.

A good illustration of the type of problem encountered and the commonsense solution submitted concerns one of the fundamentals of the code, the pipe wall thickness formula. Those familiar with the theories of failure will appreciate that there are several hypotheses currently in good repute, each of which has numerous backers. At the moment, I believe the so-called Mises-Hencky maximum energy of distortion theory seems to be considered most closely verified by experiment.

The problem of selecting a suitable formula for the code was submitted to a subgroup for consideration. After a long period of study in the subgroup, where the various theories of failure were considered, the committee reported a majority in favor of the Barlow formula, with the Lamé formula as an alternate for the thicker wall pipes. While neither the Barlow nor Lamé formula is theoretically correct in accordance with the maximum energy of distortion theory, they both have proved to be safe in practice and to produce results which are sufficiently close to the theoretical to be acceptable.

Wording the Code

It should be remembered that this code is a practical guide presenting minimum safety requirements for pressure piping, and expressed in language which is intelligible to all who design, manufacture, and erect piping systems. Particular effort was made, therefore, to exclude complicated formulas or requirements which would be impossible for the ordinary user to apply, or for the inspector to interpret.

Realizing that the code will be referred to as the authority in the field of pressure piping, the subcommittee and the editing committee have made particular effort to select the language and wording of the requirements so that the intended meaning is explicit. For example, considerable correspondence was exchanged regarding the note to the pipe wall thickness formula which permits using the actual measured wall thickness and outside diameter in calculating the maximum internal service pressure, before a mutually acceptable wording could be agreed upon. In using the code, I believe it is safe to assume that the wording of each requirement means exactly what it says. Should any requirement be considered ambiguous, however, subcommittee I is authorized to issue whatever interpretations are necessary.

Another particularly tough problem has involved the assignment of efficiencies for welded pipe. This has been a prolific source of argument ever since the project for a piping code was first initiated. The efficiencies for pipe as agreed upon for the 1935 edition of the code were as follows:

Seamless	1.00
Fusion welded, ASME paragraph U68 rules	0.90
Resistance welded	0.85
Fusion welded, ordinary	0.80
Forge welded	0.80
Lap welded	0.80
Butt welded	0.60

These values, while arbitrary to some extent, were substantiated by bursting tests carried out on full length, full size pieces of pipe.

When this revision of the code was initiated, one of the first items called for was a review of the weld efficiencies of pipe. Certain of the members of the committee were dissatisfied in particular with the efficiency assigned to electric resistance welded pipe. After much nonproductive discussion in a main committee meeting, the matter was assigned to a subgroup for further study. After prolonged discussion in the subgroup, it was agreed to recommend that a note as follows be included in each section where resistance welded pipe was permitted:

> For applications where the temperature does not exceed 450 F, and where pipe furnished under this classification is subjected to supplemental tests and/or heat treatments as agreed to by the supplier and purchaser, and whereby such supplemental tests and/or heat treatments demonstrate the strength characteristics of the weld to be equal to the plate, the *S* values equal to the corresponding seamless grades may be used.

When the code had reached its final stages and had been submitted to letter ballot of the entire sectional committee, representatives of the American Iron and Steel Institute (a trade association of the steel industry) voted negatively on the code because the above wording was not satisfactory. In committee work as well as in Congress, if you don't like the setup you can vote against it. In committee work, we make an effort to effect a revision which will be acceptable to those who favor the status quo, and those who wish a change. In this case, a new wording was submitted and a separate letter ballot was sent out to the committee for approval. This example is cited to indicate why it takes time to effect a revision of an important code.

Pot Shots

The uninitiated might conclude that the end was in sight after a preliminary draft of the code had been prepared. On the contrary, this just means that the material now is in draft form at which everyone can take a pot shot. Consideration of the many suggestions submitted is undertaken, and perhaps a new draft is prepared, which is submitted to the same procedure. This may continue for a matter of a year or so before each subcommittee feels that it has attained a satisfactory meeting of minds of the majority of its members. The subcommittee then votes for submission of its section to the committee as a whole.

While in so far as possible, progress on a project of this nature is carried out by correspondence, it is necessary that subcommittee meetings be held frequently and main committee meetings occasionally to review the progress of the work, to consider problems that cannot be solved by correspondence, and to guide the future work of the committees into the proper channels. These meetings, more often than not, last from early until late. It is hard, however, to convince the folks at home of this. "Have a good time," they say, as if one were leaving for a vacation.

At this point in committee procedure, the editing committee attempts to correlate the several sections in an effort to unify the terminology, correct editorial errors, work for uniform continuity of subject matter in each section, and in general to provide an editorially acceptable piece of work. This, in itself, is no small job. The entire code is then set in printer's proof form, re-examined by each subcommittee and re-edited by the editing committee, and a second printer's proof is prepared. The latter may then be ready to submit to letter ballot vote of the sectional committee. This procedure never fails to produce a large volume of comment. In this instance, over 60 pages of assembled comments were forthcoming, which were considered in preparing the final draft of the code.

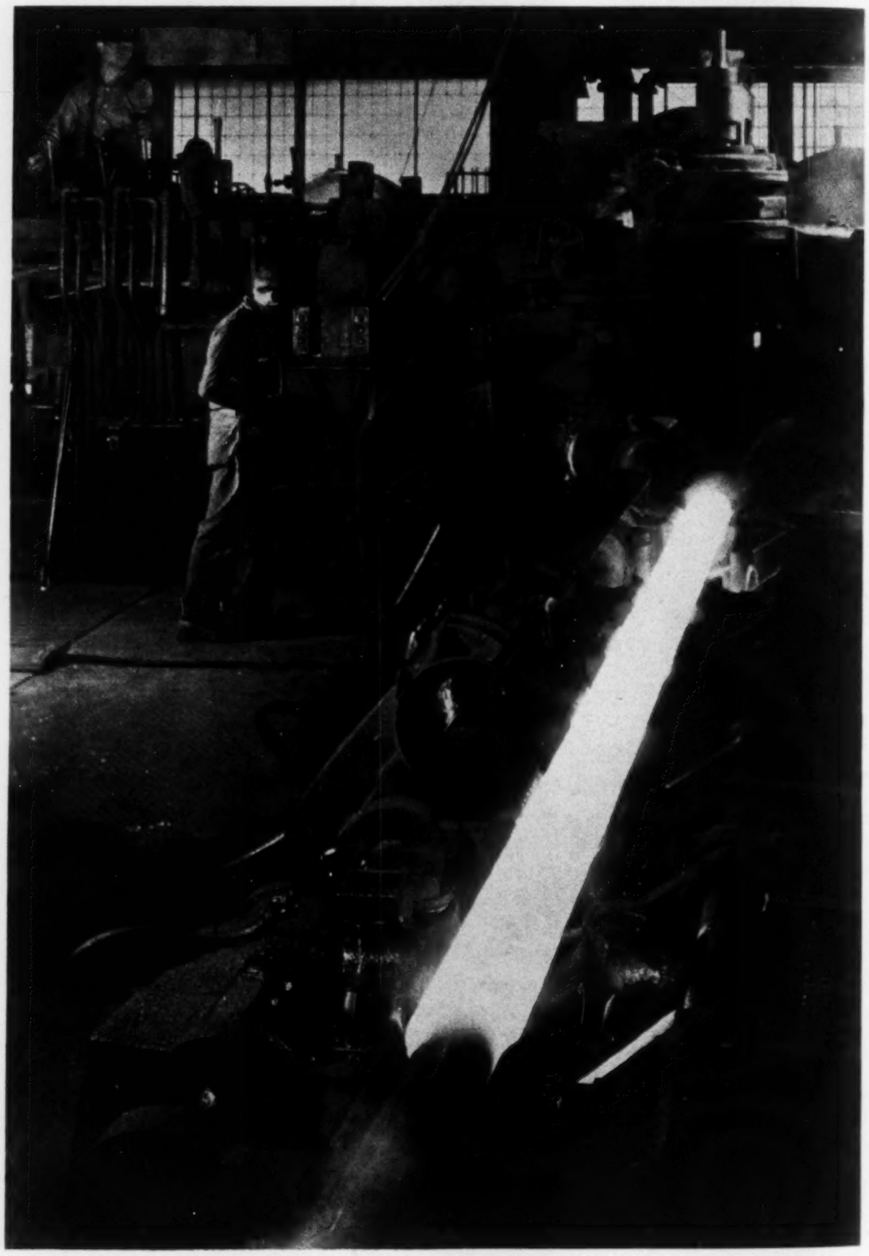
Interpretations

Provision has been made for interpreting the requirements of the code—should it be necessary—by subcommittee I on plan, scope, and editing. The personnel of this committee consists of the chairman and secretary of the main committee together with the chairmen of each of the subcommittees. It is planned that this committee will function in a manner somewhat similar to the ASME boiler code committee, which issues interpretations of the boiler code as required.

The completed work therefore represents the combined effort (not to mention time and money) of the more than 100 representatives and their associates who have labored mightily for over four years—and, it is hoped, have brought forth a revised edition of the piping code which will prove useful to all who have occasion to refer to it.

A phase in the manufacture of pipe for pressure piping

Photo courtesy National Tube Co.



Air in Air Raid Shelters

Excerpts from the Brackett lecture delivered at Princeton University by Walter L. Fleisher, consulting engineer, president of Air & Refrigeration Corp., and member of HPAC's board of consulting & contributing editors

George Stewart, in his novel, "Storm", says: "As a crab moves in the ocean bottom, but is of the water, so man rests his feet upon the earth, but lives in the air. Man thinks of the crab as a water animal; illogically and curiously, he calls himself a creature of the land."

The whole history of air raid shelters is built around air, because although we can live for long periods without food or water and can suffer from cold and some heat for hours at a time, a minute or two without air encompasses our death. Not only that, but we breathe whatever substitute is presented in place of air, and if that substitute is lethal we are the victims of our own necessity.

It was rather with a knowledge of these facts that Europe—and, for that matter, Asia—prepared for a savage war. In other words, air raid shelters were designed for protection against poison gas.

Another point in their provisions for defense must be carefully analyzed before we follow the European design in our own conception of needed protection. England, for instance, designed for 3 to 12 hr attacks, continuous and repeated day after day. This requires close bases to the attacking point. The fall of the Low Countries and of France made such attacks possible.

My remarks so far are preliminary to my saying that though we in our cities may be attacked, the same sort of defense that Europe has had to guard against during the past four years will not have to be visualized.

Let me tell you a few of the things the English did not know when I went to Europe in 1938 and which, when I brought them to their attention, I always figured (not too seriously) had something to do with the Munich appeasement. Shelters sealed as tightly as they could be were contemplated, with air introduced by forced means through gas eliminators, which were nothing but glorified gas masks of large size.

I was horrified to find that the plans for ventilating these shelters were based on 50 cfh of air per person. I am sure that everyone here knows about the metabolic output of human beings; that under the best conditions of calm, sedentary occupation the heat given out by a human being in an hour amounts to 400 Btu; that 1 Btu can raise the temperature of a cubic foot of air 55 deg. I am not going through the mathematics of showing that theoretically 50 cfh of air per person should give a rise in temperature of between 200 and 300 F, because I am sure everyone here has had enough mathematics or thermodynamics to figure that out for himself. Of course this theoretical rise in temperature is ridiculous, because a certain amount of heat would be dissipated through the enclosure and walls, and anyway most of the people would die and stop producing heat long before its ultimate effects could be experienced.

In England, France and Belgium, in Germany and in Chung King, shelters were synonymous with caves or something underground. That is because from time immemorial, when there was danger, people ran into caves to escape their enemies or to escape the elements. In 1756, in the Black Hole of Calcutta, 136* of 146 men died in 20 hr from being crowded into a cell which had only one opening and where the heat and humidity became so great that the people could not get rid of their normal metabolic heat output. For many years this catastrophe was thought to be due to lack of oxygen, but it has been definitely decided that the heat and humidity were almost entirely responsible for the deaths. The reason that England decided to go underground for their shelters was due to the fact that the earth temperature, somewhere in the lower 50 deg, would absorb the heat from the people, and the temperature of the walls would condense out the moisture, making it possible for the occupants to exist for a longer time in underground shelters without ventilation than would be possible in any superstructures that might be built.

There is another thing that occurs when people are put into confined spaces when inadequate ventilation is provided. The carbon dioxide content of the air rises, due to the fact that an average adult requires about 18 cfh of air for breathing and gives off about 4 per cent of CO_2 by volume, or about 0.7 cfh of carbon dioxide.

Garage Shelters

While I was in London, I recognized the conditions against which they were struggling, and I suggested to some prominent people, who had a good bit to say, that they should forget all about caves, tear down some of the buildings in their congested districts and build multistory garages. These garages would satisfy two requirements:

1) They would get the vehicles off the streets so that the fire department could function properly.

2) If they were ramped garages, such as we have built in some of our cities, the people, without climbing stairs, could get to a point 30 ft above the street level, which would make them immune from poison gas attacks (most poison gases staying below the 30 ft level) and from fragments. As the garages could be windowless, there would be no shattering of glass. They would be easy to ventilate, because the ventilation problem would not be one of ventilating against gas, and larger air quantities could be introduced, which would be part of the ordinary venti-

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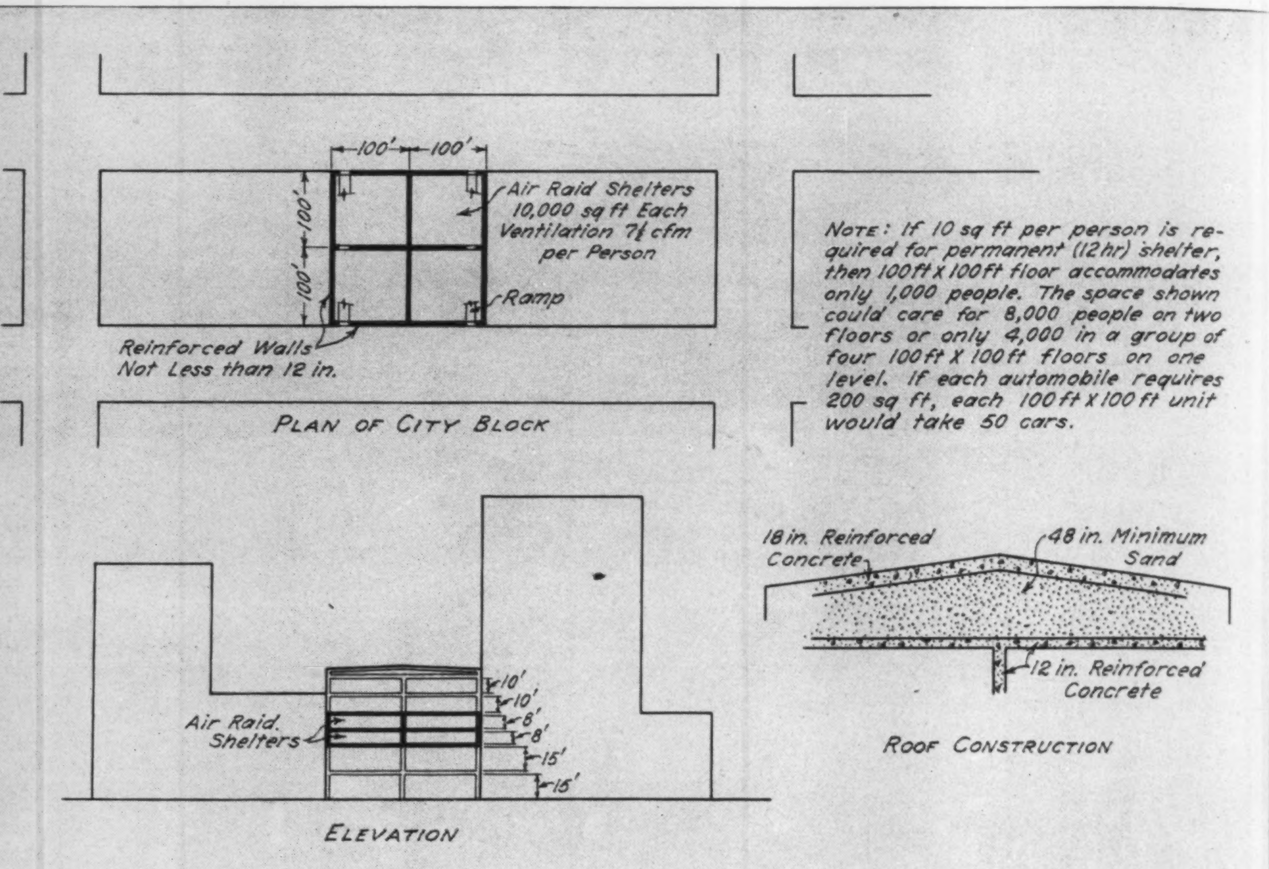
*Some accounts say 123.—Ed.


100'
100'
100'
100'
Air Raid Shelters 10,000 sq ft Each Ventilation 7½ cfm per Person
Ramp
Reinforced Walls Not Less than 12 in.
PLAN OF CITY BLOCK
18 in. Reinforced Concrete
48 in. Minimum Sand
12 in. Reinforced Concrete
ROOF CONSTRUCTION
Air Raid Shelters
10'
10'
8'
8'
15'
15'
ELEVATION

NOTE: If 10 sq ft per person is required for permanent (12 hr) shelter, then 100 ft x 100 ft floor accommodates only 1,000 people. The space shown could care for 8,000 people on two floors or only 4,000 in a group of four 100 ft x 100 ft floors on one level. If each automobile requires 200 sq ft, each 100 ft x 100 ft unit would take 50 cars.

PROPOSED COMBINATION CITY GARAGE
AND AIR RAID SHELTER

SUGGESTED AND DESIGNED BY WALTER L. FLEISHER, CONSULTING ENGINEER

lation system for such a garage. Above the shelter floor, I contemplated additional floors, properly reinforced with concrete, sand and other floors to break the impact of almost any bomb, including high explosives. These structures would have an economic value not only during the war but for many years afterwards.

This type of protection, of course, is suitable and adequate only for the congested urban districts. I have said over and over again that the Anderson shelter, which England provided for its isolated dwellings, was safe enough. This shelter, was only a corrugated iron canopy, buried in the ground and often covered with flowers and vegetables, and a protection from one's own house rather than from bombs. Elaborate structures in country districts are unnecessary and a drain on our resources in these times.

I have a feeling that whatever attacks we have, particularly on our cities and industrial centers, will be sharp but short, and the idea of living for long periods of time, day after day, in such shelters is negligible. But whatever we do and wherever we go, we must remember that people give off heat and moisture and carbon dioxide, and that any of these or all of these together, if they are beyond the endurance of the average human being, may cause more deaths than all the bombs the enemy may drop.

Proposed multistory air raid shelter garage. The air raid shelter floors should have access doors from one section to another, so that in case of any blockage in one section, the occupants could emerge from another; and would be equipped with blow-out sections, so that in the event of an explosion the expansion of the gases could be taken care of by these emergency openings. Toilet facilities are essential, as well as facilities for bunking people overnight. The whole idea is to make the population feel so secure in the idea that the municipality is taking care of them, that they will return regularly to their work rather than stay at home. The air raid floors would have to be equipped with emergency lighting systems, such as a small diesel generator, because the psychological effect of darkness is very unfortunate, and the main cables might go out during an air raid

L-107 REPLACES MARCH 24 FREEZE ORDER

Shipments of certain types of space heating equipment were limited last month by the WPB to orders of the Army, Navy, Maritime Commission, and Coast Guard. This action, embodied in limitation order L-107, replaced a telegraphic "freeze order," issued on March 24.

L-107 covers extended surface heating equipment, including unit heaters, unit ventilators, blast heating coils, convectors, and winter air conditioners. The order prohibits delivery except for the military or naval services noted above, unless expressly authorized on form PD-412a by the director of industry operations. It does not prevent delivery of electric motors or controls necessary for repair purposes, or of any repaired heat transfer element.

All persons affected must file form PD-467 with the WPB before the 15th of each month.

HEATING, PIPING & AIR CONDITIONING DATA SHEET

INSULATION OF COLD PIPE LINES

THE INDUSTRIAL Mineral Wool Institute has adopted and released specifications for the use of mineral wool in low temperature installations. Following is the section of these specifications pertaining to the application of mineral wool felt on cold pipe lines:

I. Recommended Thickness—The following table shows the minimum recommended thickness of mineral wool felt for service at various operating temperatures:

Operating Temperatures, F	Total Built-up Insulation Thickness, In.
45 to 15	2
15 to — 5	3
— 5 to —20	4
—20 to —40	5
—40 to —60	6

II. Preparation of Surface—All surfaces shall be thoroughly cleaned and dried before insulation is applied and once the installation has begun, the system shall not be put into operation until the application has been completed. Pipes and other equipment to be insulated, shall be re-located if necessary to provide an uninterrupted clearance around the finished insulation of at least 4 in. in all directions. Low temperature pipes shall not be located adjacent to heated surfaces. Pipe covering which may be subjected to abrasion shall be suitably protected.

III. Application of Mineral Wool Felt—The surface to be insulated shall first be wrapped with a layer of 15 lb saturated rag felt, mopped with hot (180 to 200 F melting point) asphalt, and lapped at least 3 in. at all joints. The initial layer of 1 in. thick mineral wool felt shall then be applied circumferentially around the pipe, securely tying it with jute twine wrapped spirally on 1 in. centers. A layer of 15 lb rag felt shall be mopped with hot asphalt and applied with 3 in. laps over the first layer of insulation. Subsequent 1 in. thicknesses of mineral wool felt shall then be applied and sealed in a similar manner until built up to the total specified thickness. Over the final layer of mineral wool felt, two layers of 15 lb saturated rag felt shall be applied with 3 in. laps, mopping each layer separately with hot asphalt.

IV. Fittings—Wherever the pipe is interrupted by fittings, valves or flanges (and also at every 15 to 20 ft of straight pipe) the pipe insulation shall be carefully sealed off as shown in the accompanying sketch. Each layer of mineral wool felt shall be sealed at these points by carrying the 15 lb rag felt down to the previously applied layer of felt, *not* down to the bare pipe in each case. All fittings shall be insulated separately, applying an equivalent thickness of insulation in the same manner as on adjacent straight piping. Where pipe and fitting insulation meet, the 15 lb rag felt applied on the fittings shall be well sealed to the outer layer of 15 lb rag felt on the pipe insulation.

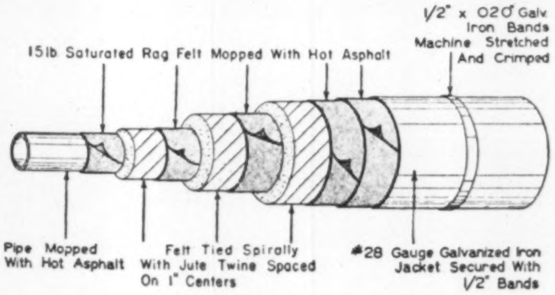
Hangers shall be insulated separately, the same as fittings, running the insulation along the supporting hanger rod a distance not less than 12 in. beyond the adjacent pipe insulation.

V. Finish on Straight Piping—After the second outer layer of 15 lb rag felt has been mopped with asphalt and applied, the insulation shall be covered with a jacket of No. 28 ga galvanized sheet metal or weatherproof roofing felt. Separate pieces of sheet metal used shall be lapped 3 in. against the weather at all joints and permanently secured in place with ½ in. galvanized iron bands spaced on 8 in. to 12 in. centers, machine stretched and crimped. If roofing felt is used as a jacket, all joints shall be lapped as above and the jacket permanently secured in place with copper or soft galvanized wires, spaced on 6 to 8 in. centers.

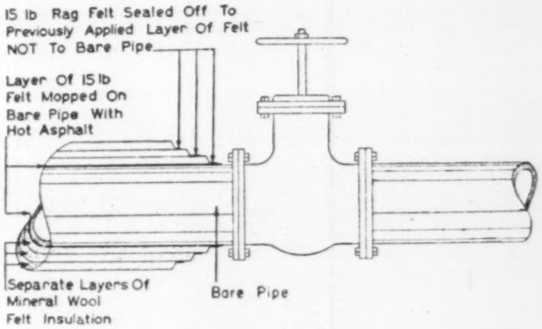
VI. Finish on Valves and Fittings—A layer of 1 in. galvanized wire mesh netting shall be tightly stretched over the final layer of 15 lb rag felt and secured in place with No. 18 ga wire. Emulsified asphalt weatherproofing shall then be applied at the rate of 2 lb per sq ft of area (approximately ¼ in. thick wet), trowelling it well into the wire mesh netting.

VII. Finish on Valves and Fittings (Alternate)—A layer of rosin sized sheathing paper shall be applied over the final layer of 15 lb rag felt, lapping it 3 in. at all joints and sealing the laps with asphalt. A jacket of 8 oz canvas shall then be sewed over the rosin sized paper, spacing the stitches no less than three to the inch located where least visible.

VIII. Painting — Insulation finished with weatherproofing shall be painted with aluminum or asphalt base paint only when specified. Insulation finished with 8 oz canvas shall be given a coat of glue sizing followed by two coats of white lead in oil tinted to color selected by the purchaser.


15lb Saturated Rag Felt Mopped With Hot Asphalt
1/2" x 020" Galv Iron Bands Machine Stretched And Crimped
Pipe Mopped With Hot Asphalt
Felt Tied Spirally With Jute Twine Spaced On 1" Centers
#28 Gauge Galvanized Iron Jacket Secured With 1/2" Bands

BUILT UP PIPE INSULATION USING MINERAL WOOL FELT


15 lb Rag Felt Sealed Off To Previously Applied Layer Of Felt NOT To Bare Pipe
Layer Of 15 lb Felt Mopped On Bare Pipe With Hot Asphalt
Separate Layers Of Mineral Wool Felt Insulation
Bare Pipe

METHOD OF SEALING OFF
PIPE INSULATION AT VALVES, FLANGES AND FITTINGS

"Open for Discussion"

SIZING VENT PIPING FOR SAFETY VALVES

THE EDITOR—

I was interested in Mr. Petersen's article on the Watts Bar steam plant published in the May HPAC.

I have not tried to analyze in detail the difficulty with the safety valve vents which Mr. Petersen described, but one thought concerning it comes to mind immediately. This is that whereas Mr. Petersen attributed the trouble to excessive friction in the vent piping because of the more dense fluid, it is of at least equal importance to question the velocity pressure with which the more dense fluid leaves the safety valve. Because water, or extremely wet steam, would flow through the safety valve much less freely than dry steam, I seriously doubt that the velocity head with which the wet mixture leaves the valve is anywhere near the value that would be obtained with dry or superheated steam. Without data for the particular problem, it is difficult to arrive at a definite answer, but it would seem that the combined effects of excessive moisture in the valve and in the vent pipe could easily account for the blowback that was experienced.

It is a source of much satisfaction to have Mr. Petersen go on to say that the vents functioned properly when the valves were definitely blowing steam. This should be the case under all normal circumstances.

With correctly sized umbrella fitted vents there frequently is a small initial puff of steam when the valve pops open, but the proper flow is established quickly and no further blowback occurs.—MAX W. BENJAMIN, mechanical engineer.

THE EDITOR—

I am very much interested in Mr. Benjamin's comments.

You will recall that in my article I said that the vent piping functioned satisfactorily when the safety valves were blown *by hand* when there was approximately a 35,000 kw load on the generator. Since the article was written the plant has been put on the line and two units are operating at full load or above. Recently we lost the load on one unit and both safety valves popped (the shutoff valve ahead of the power control valve was closed so it did not blow). Enough steam was blown out of the umbrella fittings, when this occurred, to shower all the equipment in the immediate neighborhood and form dense clouds of vapor throughout several floors of the plant. This condition lasted for quite some time.

This problem was discussed at length with the valve manufacturer, and the final result was that we are installing separate vent pipes for the power control valve, and for the combined vent from the two safety valves. We installed an expansion joint in place of each umbrella fitting from the escape valves, making in effect a closed vent to atmosphere. We also raised the power control valve about 80 ft, thus eliminating 80 ft of vent piping to cause back pressure.

I am not prepared to say what was the cause of this excessive blowback of steam. The valve manufacturer points out that considerably more steam is released from the safety valve when it pops automatically than when it is lifted by hand. This, of course, accounts for the initial satisfactory functioning of the vent manifold mentioned in my article.

This letter is not meant as a refutation of Mr. Benjamin's method of calculating safety valve exhaust vent lines. It is quite probable that several factors of design combined to give the excessive blowback. In calculating the resistance in the manifold several assumptions were necessary due to its relatively complicated structure. It is possible that too low values were assigned to points of abrupt change in direction of flow, or, it may be that the umbrella fittings were too small. Mr. Benjamin's theory [presented in the October, 1941, HPAC] seems sound and we shall continue to use it in the design of safety valve vent pipes.—H. J. PETERSEN, principal mechanical engineer, Tennessee Valley Authority.

Readers of HEATING, PIPING & AIR CONDITIONING are invited to contribute their views to "Open for Discussion," which will appear from time to time.

STEEL FOR BATTLESHIPS BY CONDITIONING BLAST FURNACES

Enough extra steel every four months to build a battleship can be obtained by air conditioning a single blast furnace, Russell V. D. Dunne, Carrier Corp. engineer, told the American Society of Refrigerating Engineers at their meeting last month. He estimated that provision of "dry blast" air conditioning for 50 blast furnaces would result in production increases of at least 10 per cent.

"If the war should last from three to five years, as many experts believe it will, air conditioning 50 existing blast furnaces would give the United States additional steel sufficient for 60,000 thirty ton tanks or 900 destroyers," Mr. Dunne asserted.

"The cost of 'dry blast' equipment for these furnaces would be from $6,500,000 to $8,000,000 and the equipment would call for 3000 tons of steel. Four to eight months would be required to provide the equipment. In order to obtain an increase in production equivalent to that produced by air conditioning 50 furnaces, it would be necessary to build five new blast furnaces. This would call for a capital expenditure of from $15,000,000 to $20,000,000 and would require from 18 to 24 months to build," he said.

"In addition," Mr. Dunne pointed out, "the amount of steel required to build new furnaces would approximate 45,000 tons."

"Dry blast," which removes tons of water from the tons of air pumped daily into a blast furnace, also results in a sizable reduction in coke consumption and produces pig iron of greater uniformity," Mr. Dunne asserted.

How to Protect Equipment Against FAILURE OF FLOW

By William Goodman

SUMMARY — In war production plants where water is being chilled by refrigeration compressors for process work or air conditioning service, protection against the failure of flow of water is vitally necessary. Failure of flow may cause a disastrous freeze-up. . . . In this article (the first part of which appeared last month, and which is concluded this month) the use of orifices and differential pressure electric switches is shown to be a superior method for stopping the compressor in case the flow of water ceases. . . . The author is consulting engineer with the Trane Co., and a member of HPAC's board of consulting and contributing editors

New Data Allow Easy Selection of Orifices for Use With Differential Pressure Switches to Guard Against Damage from Flow Failure in Chilled Water and other Piping Systems

THE CONSTRUCTION of an orifice which is satisfactory for operation with a differential pressure switch is illustrated in Fig. 1. The various dimensions corresponding to this illustration are given in Table 4. The dimension D_o is, of course, the diameter of the orifice selected by means of Table 2, which was published last month. The outside diameter of the orifice, *D,* listed in Table 4, is the correct size to fit inside the bolt circle of 125 lb standard cast iron companion flanges or of 150 lb standard steel welding neck flanges.

Orifices should be constructed of nonrusting metals such as brass, bronze, monel, or stainless steel. The thickness of the edge of the orifice, *b,* of Fig. 1 may be found in Table 4. Although the thickness may be less than the dimension shown in Table 4, it should not be greater. The upstream edge of the orifice should be sharp and square; not rounded. Needless to say, the edges should be free from burrs.

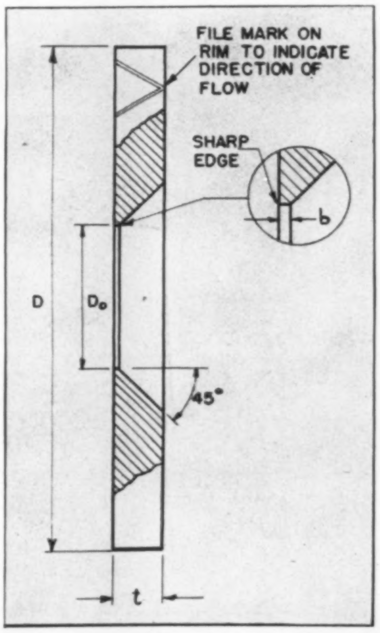
After the orifice is finished, it is a wise plan to file a V or an arrow on the rim to indicate the direction of flow through the orifice. This elementary precaution may sometimes avoid a great deal of trouble and annoyance after the orifice is installed in the pipe line.

Orifice Installation

An orifice will not give accurate results unless it is preceded and followed by a length of straight pipe without any joints. The minimum lengths of pipe that are required ahead of an orifice with different pipe fittings may be found in Tables 7 to 11 inclusive. Thus if two elbows in the same plane precede a 2 in. orifice in a 3 in. pipe, Table 8 shows that a 4 ft, 1 in. length of straight pipe is required ahead of the orifice. Notice that two elbows ahead of an orifice as in Table 8 require a greater length of pipe than a single elbow as in Table 7. The reason for this is that the two elbows produce a greater disturbance in the flow and consequently a greater length of pipe is required before a normal flow pattern is re-established within the pipe.

The lengths of pipe required when valves precede the orifice are shown in Table 11. This table may be used for either gate valves or globe valves. Wherever possible, it is best to avoid installing valves ahead of the orifice; valves should preferably be installed on the downstream side. When gate valves are installed ahead of an orifice, they must be wide open at all times.

Fig. 1—An approved type of orifice. For dimensions, see Table 4



The more common fitting arrangements are shown in Tables 7 to 11. The lengths required for fittings not shown may be approximated from these tables. For example, the length of straight pipe required when a tee precedes the orifice is about the same as for an elbow.

Table 6 shows the length of straight pipe required after an orifice. Although an elbow is shown following the orifice in the illustration at the top of Table 6, the pipe lengths shown in this table may also be used if a tee or a valve follows the orifice. The downstream lengths of pipe shown in Table 6 are suitable where the direction of flow is changed only once. In some installations, the orifice is followed by a double elbow similar to the double elbow shown ahead of the orifice illustrated in Table 8. If the orifice is followed by a double elbow, the downstream length of pipe should be equal to the upstream length shown in Table 8.

When making up a pair of companion flanges for an orifice, the pipe should be screwed into the flange until it is flush with the face

TABLE 4

ORIFICE DIMENSIONS

See Fig. 1

Nominal Pipe Size Inches	Outside Diameter Inches D	Thickness Inches t	Maximum Thickness of Orifice Edge Inches b
2	4	1/8	0.04
2-1/2	4-3/4	1/8	.05
3	5-1/4	1/8	.06
4	6-3/4	1/8	.08
5	7-5/8	1/8	.10
6	8-5/8	1/8	.12
8	10-7/8	1/4	.16
10	13-1/4	1/4	.20
12	16	1/4	.24

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of the flange. The hole in the gasket should have a diameter slightly greater than the inside diameter of the pipe so that if the gasket is not centered exactly, it will in no case project beyond the pipe wall. Fig. 2 shows the way the pipe, gasket, and orifice should fit together. The edge of the pipe is flush with the edge of the flange and touches the gasket. The gasket is undercut slightly so that it does not extend beyond the pipe wall.

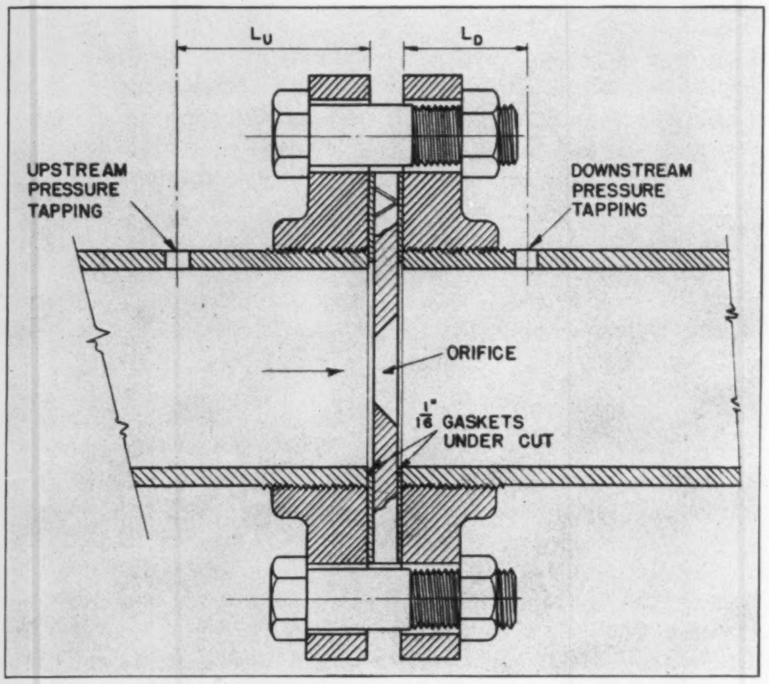

L_U
L_D
UPSTREAM PRESSURE TAPPING
DOWNSTREAM PRESSURE TAPPING
ORIFICE
$\frac{1}{16}$" GASKETS UNDER CUT

Fig. 2—Method of installing orifice between flanges, and distances between flanges and pressure tappings. Dimensions for locating pressure tappings are given in Table 5

The piping system should be checked to be sure that the orifice will not act as a dam and hold up water when the piping system is drained. If there is any possibility of the orifice acting as a dam, drain connections should be provided on both sides of the orifice. However, the drain connections should not be near the orifice; they should be installed beyond the lengths of straight pipe specified in Tables 6 to 11.

Be sure when installing the orifice between the flanges that the mark on the rim points in the same direction as the flow.

Pressure Tappings

The location of the pressure tappings is critical and great care should be exercised in drilling them correctly. The location of the upstream and downstream tappings is illustrated in Fig. 2. The dimensions noted in Fig. 2 are listed in Table 5. L_U is the distance between the face of the upstream flange and the center line of the upstream pressure tapping, and L_D is the distance from the face of the downstream flange to the center line of the downstream pressure tapping. The dimensions shown in Fig. 5 are based on the assumption that the orifice will be installed between $\frac{1}{16}$ in. thick gaskets.

Pressure tappings in horizontal pipe lines should not be installed either at the top or the bottom of the pipe. The proper location for the tappings is through the side of the pipe as shown in Fig. 3. Top connections may become airbound, and bottom connections may become plugged with dirt.

The pressure tappings through the wall of the pipe should be ¼ in. in diameter or smaller. Inside the pipe the edge of the ¼ in. hole should be free from burrs and slightly rounded. A good method of making connections to these pressure tappings is to weld a ¼ in. pipe nipple about 6 in. long to the outside of the pipe over the ¼ in. hole, as shown in Fig. 3. Only one end of the ¼ in. nipple should be threaded. The unthreaded end should be filed with a halfround file to fit the contour of the pipe. The ¼ in. hole through the wall of the pipe should be drilled before the nipple is welded to the pipe. Flare couplings are used to connect copper tubing to the ¼ in. pipe nipples.

TABLE 5

LOCATION OF PRESSURE TAPPINGS
(Radius Taps)

Inches

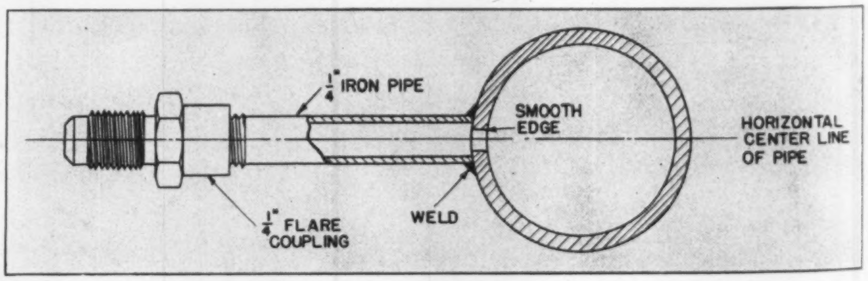
See Fig. 2

Nominal Pipe Size Inches	Upstream Tapping Inches	Downstream Tapping Inches
	L_U	L_D
2	2.00	0.97
2-1/2	2.41	1.17
3	3.01	1.47
4	3.96	1.95
5	4.98	2.46
6	6.00	2.97
8	8.01	3.97
10	10.07	5.01
12	12.03	5.98

Note: Dimensions in this table based on orifice plate being installed between 1/16 in. thick gaskets.

On pipes 8 in. and larger, a hole can be drilled and tapped for ⅛ in. iron pipe, and a ⅛ in. nipple screwed into the pipe. However, the ⅛ in. nipple should not extend through the wall of the pipe. The nipple should either be flush on the inside or, better still, slightly recessed. Avoidance of any projection inside the pipe is highly important if erroneous pressure differences are to be avoided. To make sure that the threaded end is cut off

Fig. 3—Pressure tappings should be drilled through side of pipe, not through the top or bottom. Top connections may become airbound, and bottom connections may become plugged with dirt



at the right point so that it will not project into the pipe, it is well to first try the nipple in a short length of pipe where the projection can easily be checked.

For all pipe sizes except 2, 2½, and 3 in., the downstream pressure tapping will clear the hub of the flange. For 2, 2½, and 3 in. pipe, the downstream tapping is partially underneath the hub of the flange. Consequently, when installing orifices in 2, 2½, and 3 in. pipe, it is better to use welding neck companion flanges. In such flanges, the pressure tapping can be drilled directly through the neck of the flange.

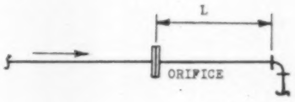


TABLE 6

MINIMUM STRAIGHT LENGTH OF PIPE REQUIRED ON DOWNSTREAM SIDE OF ORIFICE

Nominal Pipe Size Inches	Pipe Length L_D
2	0'-10"
2-1/2	1 - 0
3	1 - 3
4	1 - 8
5	2 - 1
6	2 - 6
8	3 - 4
10	4 - 3
12	5 - 0

Note: If a double elbow is used on downstream side of orifice, the downstream pipe length shall be equal to the upstream lengths shown in Table 6.

Switch Connections

Copper tubing of ¼ in. diameter is satisfactory for connecting the pressure tappings in the pipe wall to the differential pressure switch as shown in Fig. 4. Shutoff valves are desirable in case it is necessary to remove the pressure switch. A bypass is also of great convenience for adjusting the differential pressure switch and for testing its operation. By opening the bypass between the high and low sides, any desired pressure difference can be obtained across the differential pressure switch.

If shutoff valves are to be installed, the bypass has an even more important function than just permitting the adjustment and testing of the differential pressure switch, important as these functions are. The bypass also forms a very valuable safety device to prevent the differential pressure switch from becoming inoperative. Notice in Fig. 4 that a three way valve is used on the low pressure connection to the pressure switch. If no bypass were installed and an ordinary shutoff valve used, closing the shutoff valve on the low side might keep the differential pressure switch closed regardless of how low the flow in the main pipe line fell. In fact, even if the flow ceased altogether, the differential pressure switch probably would not stop the compressor. However, with a three way valve, if the low side pressure tapping should inadvertently be closed, full high side pressure would immediately be transmitted to the low side connection of the differential pressure switch, thus immediately stopping the compressor. The use of the three way valve makes it impossible to run the compressor if the low side connection is shut off.

An ordinary backseated refrigeration shutoff valve with a gage port tapping makes an excellent three way valve for the low pressure connection to the pressure switch. On the high side connection an ordinary shutoff valve is satisfactory because if this valve should inadvertently be closed, the differential pressure switch could still stop the compressor.

Notice in Fig. 4 that two valves are shown at the bottom of the connections near the differential pressure switch. These valves are handy not only for draining and venting the connections, but also for connecting pressure gages or, better still, a manometer for adjusting and checking the operation of the differential pressure switch.

Manometers

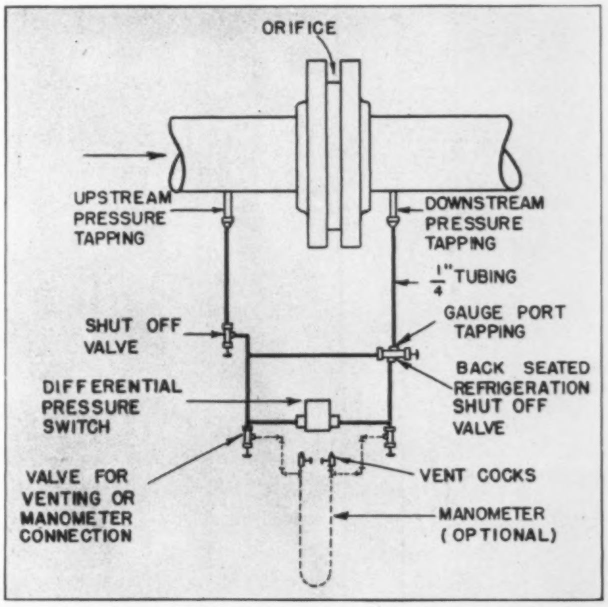
In using mercury manometers for checking the operation of differential pressure switches, it is necessary to convert pounds per square inch (psi) to inches of mercury. Ordinarily, psi multiplied by 2.036 will give in. of mercury corresponding to any given pressure difference. However, in a manometer the head of water on one leg is greater than the other and a correction must be made for this. Because of the unbalanced head of water in the manometer, the difference in mercury levels will be greater than it would be if no water were present. Consequently, instead of the value of 2.036, the value of 2.2 should be used as the multiplier, or for water, $h = 2.2\,p$.

If brine or some other liquid is flowing in the pipe line, the multiplier of 2.2 cannot be used. Instead, the following general formula can be used for any liquid:

$$h = \frac{27.71\,p}{13.6 - s}$$

Mercury manometers are more convenient and accurate to use for adjusting differential pressure switches than are two separate pressure gages. However, if a manometer is subjected to too great a pressure difference, mercury may be forced into the main water system. If mercury comes in contact with the nonferrous tubes in the water cooler or parts of the system, it will form an amalgam and cause leaks. Manufacturers can supply manometers with either special check

Fig. 4—Diagrammatic illustration showing how pressure tappings are connected to the differential pressure switch. If shutoff valves are used, a three way valve is a desirable precaution on the low pressure connection to the switch


ORIFICE
UPSTREAM PRESSURE TAPPING
DOWNSTREAM PRESSURE TAPPING
1/4" TUBING
GAUGE PORT TAPPING
SHUT OFF VALVE
BACK SEATED REFRIGERATION SHUT OFF VALVE
DIFFERENTIAL PRESSURE SWITCH
VENT COCKS
VALVE FOR VENTING OR MANOMETER CONNECTION
MANOMETER (OPTIONAL)

valves or enlarged sections at the top which will prevent mercury from leaving the instrument on too large a pressure difference.

When preparing to read a manometer, the three way valve in Fig. 4 should be so adjusted that the bypass between the high and low sides will be open, thus equalizing the pressure. After the valves ahead of the manometer have been opened, the bypass connection should be closed slowly so that the mercury will not rise too fast in the manometer. In this way the movement of the mercury can be controlled, and stopped in time if the pressure difference should be too great. However, even though the movement of the mercury can be controlled by means of the bypass

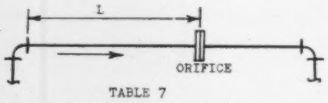


TABLE 7

MINIMUM STRAIGHT LENGTH OF PIPE REQUIRED
ON UPSTREAM SIDE OF ORIFICE

Orifice Diameter Inches	Nominal Pipe Size — Inches								
	2	2-1/2	3	4	5	6	8	10	12
5/8	1'-0"								
3/4	1 -0	1'- 3"							
7/8	1 -1	1 - 3							
1	1'-2"	1'- 3"	1'- 6"						
1-1/8	1 -4	1 - 4	1 - 6						
1-1/4	1 -7	1 - 5	1 - 7	2'- 0"					
1-3/8	2 -0	1 - 8	1 - 7	2 - 0					
1-1/2		1'-11"	1'- 9"	2'- 0"	2'- 6"				
1-5/8		2 - 5	1 -10	2 - 0	2 - 6				
1-3/4		2 -11	2 - 1	2 - 1	2 - 6				
1-7/8			2 - 4	2 - 2	2 - 6				
2			2'-10"	2'- 4"	2'- 6"	3'- 0"			
2-1/8			3 - 4	2 - 5	2 - 7	3 - 0			
2-1/4				2 - 9	2 - 8	3 - 0			
2-3/8				2 -11	2 -10	3 - 0			
2-1/2				3'- 3"	2'-11"	3'- 1"	4'- 0"		
2-5/8				3 - 9	3 - 0	3 - 2	4 - 0		
2-3/4				4 - 3	3 - 2	3 - 3	4 - 0		
2-7/8					3 - 5	3 - 4	4 - 0		
3					3'- 7"	3'- 6"	4'- 0"	5'-1"	
3-1/4					4 - 5	3 -10	4 - 0	5 -1	
3-1/2					5 - 6	4 - 3	4 - 2	5 -1	
3-3/4						5 - 0	4 - 4	5 -1	6'- 1"
4						5'-10"	4'- 8"	5'-1"	6'- 1"
4-1/4						6 -10	4 -11	5 -2	6 - 1
4-1/2							5 - 5	5 -4	6 - 1
4-3/4							5 - 9	5 -6	6 - 1
5							6'- 8"	5'-8"	6'- 2"
5-1/2							8 - 6	6 -4	6 - 5
6								7 -4	6 -11
6-1/2								9 -1	7 - 7
7								11'-2"	8'- 6"
7-1/2									10 - 0
8									11 - 9
8-1/2									13 - 8

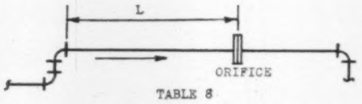


TABLE 8

MINIMUM STRAIGHT LENGTH OF PIPE REQUIRED
ON UPSTREAM SIDE OF ORIFICE

Orifice Diameter Inches	Nominal Pipe Size — Inches								
	2	2-1/2	3	4	5	6	8	10	12
5/8	1'- 6"								
3/4	1 - 6	1'- 9"							
7/8	1 - 6	1 - 9							
1	1'- 8"	1'- 9"	2'- 2"						
1-1/8	1 -11	1 -11	2 - 2						
1-1/4	2 - 4	2 - 2	2 - 2	2'-11"					
1-3/8	2 -10	2 - 5	2 - 4	2 -11					
1-1/2		2'-11"	2'- 6"	2'-11"	3'- 7"				
1-5/8		3 - 4	2 -10	2 -11	3 - 7				
1-3/4		3 -11	3 - 2	3 - 0	3 - 7				
1-7/8			3 - 7	3 - 2	3 - 7				
2			4'- 1"	3'- 4"	3'- 7"	4'-4"			
2-1/8			4 - 7	3 - 8	3 - 8	4 -4			
2-1/4				4 - 0	3 -10	4 -4			
2-3/8				4 - 4	3 -11	4 -4			
2-1/2				4'-10"	4'- 2"	4'-4"	5'- 9"		
2-5/8				5 - 4	4 - 6	4 -5	5 - 9		
2-3/4				5 -10	4 - 8	4 -7	5 - 9		
2-7/8					5 - 2	4 -9	5 - 9		
3					5'- 6"	5'-1"	5'- 9"	7'- 4"	
3-1/4					6 - 4	5 -8	5 - 9	7 - 4	
3-1/2					7 - 7	6 -5	5 -11	7 - 4	
3-3/4						7 -3	6 - 3	7 - 4	8'- 8"
4						8'-3"	6'- 9"	7'- 4"	8'- 8"
4-1/4						9 -3	7 - 5	7 - 5	8 - 8
4-1/2							8 - 1	7 - 7	8 - 8
4-3/4							8 - 9	8 - 0	8 - 8
5							9'- 8"	8'- 5"	8'- 9"
5-1/2							11 - 9	9 - 6	9 - 4
6								11 - 0	10 - 1
6-1/2								12 -10	11 - 3
7								15'- 2"	12'-10"
7-1/2									14 - 6
8									16 - 6
8-1/2									18 - 5

Note: If a double elbow is also used on the downstream side of the orifice, the downstream pipe length shall be made equal to the upstream length.

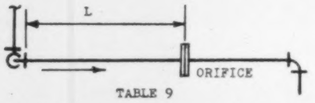


TABLE 9

MINIMUM STRAIGHT LENGTH OF PIPE REQUIRED
ON UPSTREAM SIDE OF ORIFICE

Orifice Diameter Inches	Nominal Pipe Size — Inches								
	2	2-1/2	3	4	5	6	8	10	12
5/8	2'- 9"								
3/4	2 -11	3'-4"							
7/8	3 - 2	3 -6							
1	3'- 5"	3'-8"	4'- 3"						
1-1/8	3 - 9	4 -0	4 - 5						
1-1/4	4 - 3	4 -3	4 - 8	5'- 5"					
1-3/8	4 -10	4 -8	4 -10	5 - 7					
1-1/2		5'-2"	5'- 1"	5'-10"	6'- 9"				
1-5/8		5 -9	5 - 6	6 - 0	6 -11				
1-3/4		6 -6	5 -11	6 - 4	7 - 1				
1-7/8			6 - 5	6 - 7	7 - 3				
2			7'- 0"	6'-10"	7'- 7"	8'- 4"			
2-1/8			7 - 8	7 - 3	7 - 9	8 - 6			
2-1/4				7 - 7	8 - 0	8 - 9			
2-3/8				8 - 1	8 - 2	9 - 0			
2-1/2				8'- 7"	8'- 7"	9'- 2"	10'-10"		
2-5/8				9 - 2	8 -10	9 - 5	11 - 0		
2-3/4				9 -11	9 - 3	9 - 7	11 - 3		
2-7/8					9 - 8	9 -10	11 - 6		
3					10'- 1"	10'- 4"	11'- 8"	13'- 6"	
3-1/4					11 - 3	11 - 1	12 - 1	13 -11	
3-1/2					12 - 8	11 -11	12 - 6	14 - 2	
3-3/4						12 -11	13 - 0	14 - 8	16'- 3"
4						14'- 2"	13'- 9"	15'- 1"	16'- 7"
4-1/4						15 - 7	14 - 6	15 - 8	17 - 0
4-1/2							15 - 3	16 - 1	17 - 5
4-3/4							16 - 2	16 - 7	17 -11
5							17'- 3"	17'- 0"	18'- 4"
5-1/2							19 -10	18 - 8	19 - 5
6								20 - 5	20 - 7
6-1/2								22 - 8	22 - 2
7								25'- 7"	23'- 9"
7-1/2									25 -10
8									28 - 3
8-1/2									31 - 2

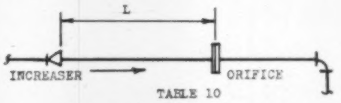


TABLE 10

MINIMUM STRAIGHT LENGTH OF PIPE REQUIRED
ON UPSTREAM SIDE OF ORIFICE

Orifice Diameter Inches	Nominal Pipe Size — Inches								
	2	2-1/2	3	4	5	6	8	10	12
5/8	1'-0"								
3/4	1 -0	1'-3"							
7/8	1 -1	1 -3							
1	1'-3"	1'-3"	1'- 6"						
1-1/8	1 -5	1 -5	1 - 6						
1-1/4	1 -8	1 -7	1 - 7	2'- 0"					
1-3/8	1 -11	1 -9	1 - 9	2 - 0					
1-1/2		2'-1"	1'-11"	2'- 0"	2'- 6"				
1-5/8		2 -3	2 - 1	2 - 1	2 - 6				
1-3/4		2 -7	2 - 4	2 - 2	2 - 6				
1-7/8			2 - 7	2 - 4	2 - 6				
2			2'- 9"	2'- 6"	2'- 7"	3'-0"			
2-1/8			3 - 1	2 - 8	2 - 8	3 -0			
2-1/4				2 -11	2 -10	3 -0			
2-3/8				3 - 2	2 -11	3 -1			
2-1/2				3'- 5"	3'- 2"	3'-2"	4'- 0"		
2-5/8				3 - 7	3 - 4	3 -3	4 - 0		
2-3/4				3 -11	3 - 5	3 -5	4 - 0		
2-7/8					3 - 9	3 -6	4 - 0		
3					3'-11"	3'-9"	4'- 0"	5'- 1"	
3-1/4					4 - 5	4 -2	4 - 1	5 - 1	
3-1/2					5 - 0	4 -7	4 - 4	5 - 1	
3-3/4						5 -1	4 - 8	5 - 1	6'- 1"
4						5'-7"	5'- 1"	5'- 2"	6'- 1"
4-1/4						6 -1	5 - 5	5 - 4	6 - 1
4-1/2							5 -10	5 - 6	6 - 1
4-3/4							6 - 4	5 -11	6 - 2
5							6'-10"	6'- 3"	6'- 4"
5-1/2							7 -10	7 - 0	6 -11
6								8 - 0	7 - 7
6-1/2								9 - 0	8 - 3
7								10'- 1"	9'- 2"
7-1/2									10 - 2
8									11 - 1
8-1/2									12 - 2

valve, a manometer should never be used without some means of preventing mercury from being forced over into the piping system. Once mercury settles in nonferrous pipes, it is almost impossible to remove it and leaks are almost certain to occur in the future.

If a manometer is permanently installed, it can be used as a flow indicator. To keep the permanent pressure loss across the orifice low when it is used for protection against flow failure, the pressure drop at the maximum flow should generally be small. Consequently in this case the difference in the mercury levels in the manometer will be small even for the maximum flow through the orifice. Nevertheless, even a small difference of, say, 4 or 5 in. can be used to obtain a fairly good idea of the quantity of liquid flowing through the orifice. For more accurate results, smaller orifices must be used with greater pressure drops which will give a larger deflection of the mercury. However, with small orifices the permanent pressure loss will be greater than with large orifices.

The data in Table 2 are more than accurate enough for designing orifices for differential pressure switches. These data are also accurate enough to enable the quantity of water flowing through the orifice to be computed under different pressure drops. However, in preparing Table 2, no account was taken of the variation in the flow coefficient of the orifice with the quantity of water flowing through it. Because of this, the flow computed by means of the data in Table 2 will not be as accurate as if the selection of the orifice flow coefficient had been based upon the Reynolds number. For further information on this point, see the ASME publication entitled Flow Measurement—1940.

Other Liquids

It frequently is desirable to protect brine circulating systems against flow failure. For this purpose, the differential pressure switch and orifice unit is also a most satisfactory method.

The orifices for brine are sized in exactly the same manner as for water. As was the case for water, the maximum pressure drop corresponding to the maximum flow can be selected from Table 1, published last month. The values in this table are independent of the specific gravity of the liquid.

Inasmuch as the values in Table 2 are also independent of the specific gravity of the liquid, this table can be used for selecting the orifice in exactly the same manner as for water. As can be seen from formula 2 (p. 349, June HPAC) the values of Z depend upon the specific gravity of the liquid. However, once the values of Z have been computed by means of formula 2, using the proper specific gravity, the orifice can then be selected from Table 2. Because for a given rate of flow the pressure drop varies directly as the specific gravity, it will be found that for a given pressure drop smaller orifices will be required when using brine and other liquids heavier than water.

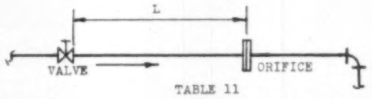


TABLE 11

MINIMUM STRAIGHT LENGTH OF PIPE REQUIRED ON UPSTREAM SIDE OF ORIFICE

Orifice Diameter Inches	Nominal Pipe Size — Inches							
	2	2-1/2	3	4	5	6	8	10	12
5/8	3'- 4"								
3/4	3 - 7	4'- 0"							
7/8	3 -10	4 - 2							
1	4'- 2"	4'- 6"	5'- 1"						
1-1/8	4 - 6	4 - 9	5 - 4						
1-1/4	5 - 2	5 - 2	5 - 8	6'- 6"					
1-3/8	6 - 0	5 - 7	5 -11	6 -10					
1-1/2		6'- 4"	6'- 3"	7'- 0"	8'- 1"				
1-5/8		7 - 1	6 - 8	7 - 3	8 - 4				
1-3/4		8 - 2	7 - 2	7 - 8	8 - 7				
1-7/8			7 -10	7 -11	8 - 9				
2			8'- 7"	8'- 4"	9'- 4"	10'- 1"			
2-1/8			9 - 7	8 - 9	9 - 6	10 - 4			
2-1/4				9 - 3	9 -10	10 - 7			
2-3/8				9 -10	10 - 0	10 - 9			
2-1/2				10'- 6"	10'- 5"	11'- 1"	13'- 1"		
2-5/8				11 - 4	10 - 9	11 - 4	13 - 5		
2-3/4				12 - 3	11 - 1	11 - 8	13 - 8		
2-7/8					11 - 9	11 -11	13 -11		
3					12'- 3"	12'- 6"	14'- 1"	16'- 2"	
3-1/4					13 -10	13 - 4	14 - 7	16 -11	
3-1/2					15 - 9	14 - 6	15 - 2	17 - 3	
3-3/4						15 -10	15 - 8	17 - 9	19'-8"
4						17'- 6"	16'- 8"	18'- 2"	20'-2"
4-1/4						19 - 6	17 - 6	18 -10	20 -8
4-1/2							18 - 4	19 - 4	21 -1
4-3/4							19 - 8	20 - 0	21 -6
5							21'- 1"	20'- 9"	22'-2"
5-1/2							24 - 7	22 - 4	23 -6
6								24 -10	25 -0
6-1/2								28 - 0	26 -6
7								31'-10"	28'-10"
7-1/2									31 -5
8									35 -0
8-1/2									38 -9

The permanent pressure loss due to the orifice when brine is being circulated can be estimated by means of the data in Table 3, which are used in exactly the same manner as for water.

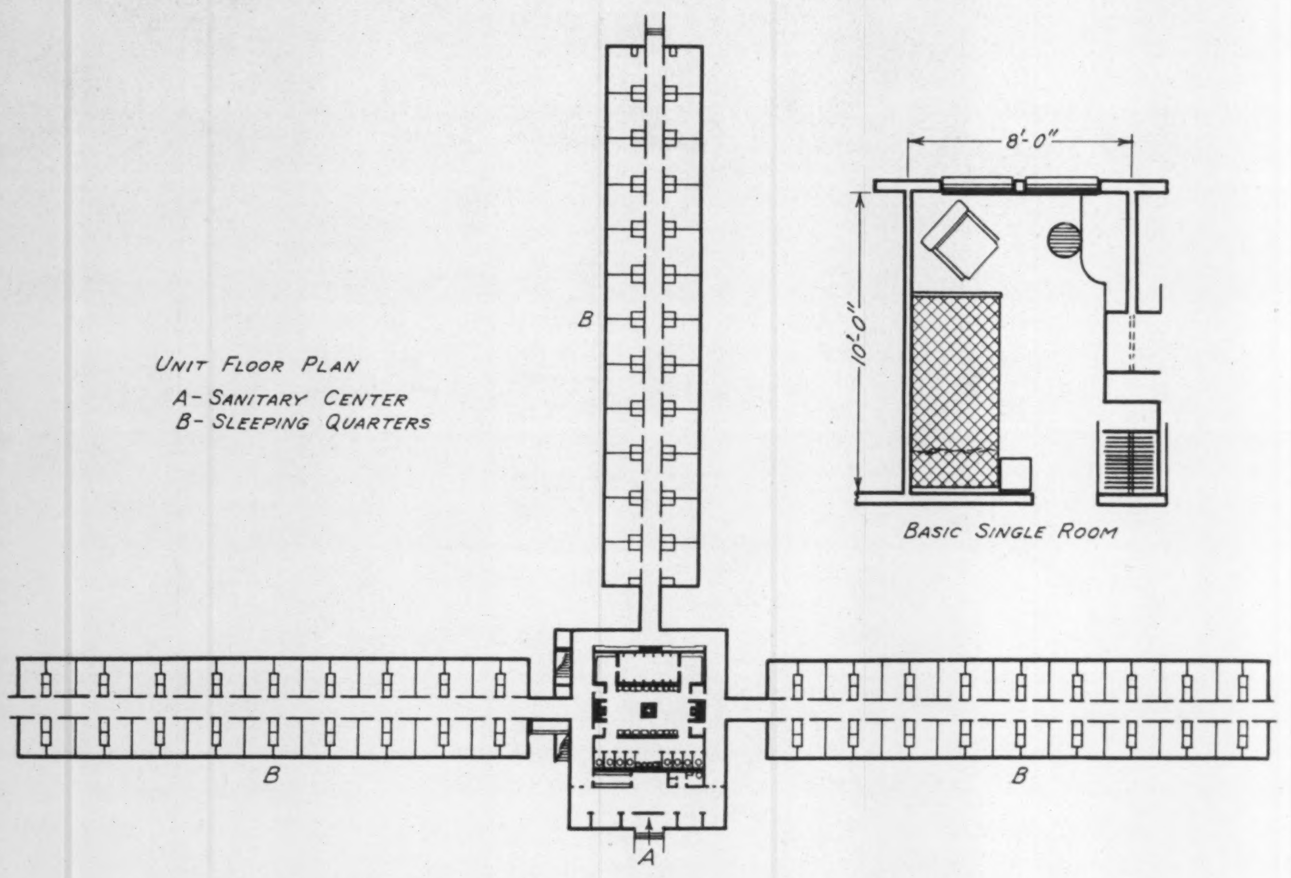
No doubt for many applications other than refrigeration and air conditioning, protection against flow failure by means of the orifice and differential pressure switch would be highly valuable. The data offered in this article will make it possible for the reader to select and install orifices and differential pressure switches for almost any liquid and any type of system likely to be encountered in practical engineering work. However, a word of caution is needed in applying Table 2. These data should not be used when the Reynolds number is less than approximately 100,000. The computation of the Reynolds number in this case is based on the diameter of the orifice; not on the diameter of the pipe.

$$(Re) = 379 \frac{W}{D_o u}$$

Reynolds numbers for most liquids commonly used in engineering applications, except oil, will usually be greater than 100,000. For data on orifice coefficients for smaller Reynolds numbers, consult the ASME publication, Flow Measurement—1940.

Symbols

b = width of orifice edge, in. See Fig. 1;
D = outside diameter of orifice, in.;
D_o = orifice diameter, in.;
h = manometer reading, in. Hg;
L_D = distance from downstream face of flange to downstream pressure tapping, in.;
L_U = distance from upstream face of flange to upstream pressure tapping, in.;
p = pressure drop across orifice for maximum flow, psi;
(Re) = Reynolds number;
s = specific gravity of fluid (for water = 1);
t = thickness of orifice plate, in.;
u = viscosity centipoises;
W = weight of liquid flowing, lb per min;
Z = orifice constant;



PRELIMINARY SKETCHES OF THE DORMITORY COMMUNITIES FOR WAR WORKERS BEING DEVELOPED BY FPHA

Dormitories for War Workers To Have Forced Warm Air Heating

FPHA Developing Duration Dormitories of Temporary Nature to Provide Essentials of Housing for Wartime Workers

"DURATION DORMITORIES," temporary in structure but providing the accommodations essential to the health and welfare of war workers, are being developed by the Federal Public Housing Authority as part of its war housing program. These self-contained dormitories are planned for a twofold purpose—(1) to promote the worker's productive efficiency and increase the output of war industry plants, and (2) reduce the use of critical war materials to a minimum and provide a solution to the growing transportation shortage resulting from the restrictions on tires, gasoline and automobiles.

Essentials for the comfort and efficiency of the workers to be provided in these dormitories include proper food to maintain physical and mental capacity, quiet sleeping rooms, medical examination and care, and recreational activities.

Each dormitory is composed of from two to four sleeping wings, with a sanitary center with lavatories and showers. Short passageways connect the sleeping wings with other parts of the dormitory.

The designs of these dormitories will be governed largely by the terrain of the sites. On level sites, a T shaped, three wing type can be used. On sloping sites, a two wing type is more adaptable.

The basement of the sanitary center contains the heating for the whole dormitory unit. Shortages of pipes, radiators, and boilers suggest the use of a forced warm air furnace, which is largely masonry with metal grates, and nonmetallic ducts, the FPHA says. Operation is most efficient if air travels equal distances in every direction; this dictates placing the furnace in the central building, and also calls for wings of equal length, maximum about 144 ft each, according to the FPHA.

Ducts from the furnace are placed above the ceiling of the corridor, supplying the rooms on either side with warm air. The entire crawl space beneath the floors is enclosed and insulated to form the return air duct. By so doing, the floors are heated and feet are kept warm.

FLOW OF SOLIDS IN PIPING

Practical Data on Pumping and Piping Sludges, Muds and Suspensions in Industrial Wartime Processes ... By H. E. Babbitt and D. H. Caldwell

SUMMARY—Knowledge of the flow characteristics of muds, sludges and solids in solution is needed in many industrial wartime processes, in oil well drilling, paper making, water treatment, sewage treatment, dredging, etc. Equations are presented here from which the pipe friction losses necessary to design the piping and select the pumping equipment may be computed for the flow of all types of such materials commonly encountered in practice. The equation for plastic flow is not original with the authors, but the equations for critical velocity and turbulent flow are presented for the first time, as far as is known . . . In a second article, measurement of the sludge constants will be discussed, and examples showing the practical applications of the equations will be given . . . The authors are, respectively, professor of sanitary engineering, and research assistant in civil engineering, at the University of Illinois.

PIPING ENGINEERS are frequently faced with the problem of designing pumping equipment and pipe lines for conveying muds, sludges or suspensions. Knowledge of the flow characteristics of muds, sludges or suspensions is needed in oil well drilling, paper making, water treatment, sewage treatment, dredging and numerous process industries. At present, pipe friction loss determinations for the flow of sludges or suspensions are based upon empirical formulas or arbitrary assumptions resulting from experience in conveying the particular sludge or suspension in question.

It has been noted by previous authors [see references numbered 1, 2, 3, 4, 5 and 6] that for a sludge, two types of flow exist similar to the laminar and turbulent states of flow in the case of a true liquid. It was noted that in laminar flow Poiseuille's equation, used for true fluids, did not apply to the flow of sludge, but that for turbulent flow the same laws applied to sludge flow as applied to true fluid flow. It was also noted that in the case of sludges, plastic or laminar flow changed to turbulent flow at a more or less clearly defined velocity, called the *critical velocity*. For the purpose of the present discussion the type of flow which occurs below the critical velocity is termed *plastic flow* and the type of flow occurring above the critical velocity is called *turbulent flow*.

There has been a marked scarcity of data in the literature dealing with the flow of sludges in pipes from which formulas applicable to all types of sludge flow problems might be deduced. Bingham [reference 6] developed an equation for plastic flow in capillary tubes. Although the flow of sludge in pipes larger than capillary tubes has been found to follow Bingham's equation over certain ranges, in turbulent flow the Bingham equation no longer applies.

The purpose here is to present equations from which pipe friction losses may be computed for the flow of all types of sludges commonly encountered in practice. The equation for plastic flow is not original with the authors, but the equations for critical velocity and for turbulent flow are presented for the first time, as far as is known. All equations have been verified by tests performed by the authors and by tests reported in the literature.

Plastic Flow

For a clear understanding of the characteristics of plastic flow it is necessary to distinguish between *viscosity* and *plasticity*. Viscosity is the measure of the resistance to flow or deformation of a *fluid*. The rate of deformation is proportional to the deforming force. Plasticity is the property of a substance which enables it to be continuously and permanently deformed in any direction, without rupture under a stress exceeding the yield value.

Fig. 1 represents viscous flow and plastic flow. Curve I represents the flow of a true liquid; the slope of the line is proportional to the coefficient of viscosity. Curve II represents the flow of a true plastic. The apparent viscosity of the plastic at any point *A* on curve II, if measured in the usual way for liquids, is proportional to the slope of the line *OA*. It is evident, therefore, that the apparent viscosity is not constant for different velocities and stresses. It is seen that two different velocities, such as *A* and *B* in the figure, correspond to entirely different viscosity lines *OA* and *OB*, the slopes of which are proportional to the apparent viscosity. It has been found in our investigation that the flow of sludge follows the type of flow illustrated by curve II. It is concluded, therefore, that sewage sludge and clay slurries are true plastics.

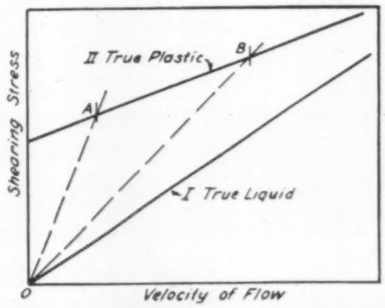


Fig. 1—Representation of viscous flow and plastic flow

In reference 1, an equation has been shown representing the flow of plastic material in a circular pipe. By rewriting it and omitting a term, this equation may be written as follows:

$$V = \frac{gD}{8\eta}\left(\tau_p - \frac{4}{3}\tau_y\right) \ldots\ldots.[1]$$

(Note—The term omitted results in an error of 5.9 per cent when $\tau_y/\tau_p = 0.5$, and 1.8 per cent when $\tau_y/\tau_p = 0.4$).

Bingham has shown [reference 6] that the coefficient of rigidity η and the yield value τ_y are independent of the characteristics of measuring apparatus and depend only upon the nature of the sludge*. Both of these facts have been corroborated in our investigation. On this basis, if a graph is plotted with the shear at the pipe wall τ_p as ordinate, against eight times the velocity divided by g times the diameter $(8V/gD)$ as abscissa, the slope of

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*A discussion of the various factors affecting the yield value and the rigidity is given in Bulletin 319 of the University of Illinois Engineering Experiment Station, entitled Laminar Flow of Sludge in Pipes with Special Reference to Sewage Sludge, by H. E. Babbitt and D. H. Caldwell.

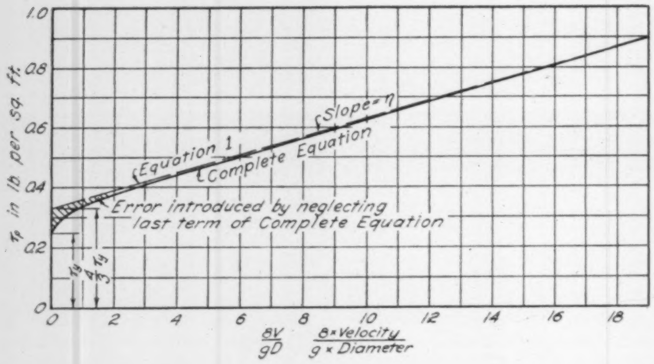


Fig. 2—Graphical representation of equation 1 and complete equation from which it is obtained

the resulting line will represent the coefficient of rigidity η; and the intercept of the line on the τ_p axis will be 4/3 the yield value. The one line, therefore, represents the flow of a given sludge in a pipe of any diameter. A graph of this type, illustrating equation 1, is shown in Fig. 2. The error in neglecting the last term of the complete equation from which equation 1 was re-written, is thus shown graphically.

For industrial piping, and with sewage sludges, clay slurries, and drilling muds as the flowing material, equation 1 will yield accurate results within the limits of experimental error in determining the yield value and the rigidity of the sludge. This equation can be written in another form which may be more convenient in certain cases, as follows:

$$\frac{H}{L} = 32\left[\frac{\tau_y}{6\rho D} + \frac{\eta V}{g\rho D^2}\right] \ldots [2]$$

Equation 2 has been checked by experiments in our investigation, and by using tests reported in the literature. It was found that the flow of sewage sludge and of clay suspensions follow closely the theoretical equation for the flow of a true plastic in circular pipe. It was also noted that both the yield value and the rigidity of the sludge are independent of the pipe diameter; this corroborates Bingham's assumptions.

Critical Velocity

The critical velocity for sludge flow will be considered as that velocity below which the friction loss is directly proportional to some power of the velocity between 1.7 and 2.0.

It has been found that by using an "apparent viscosity" in connection with the Reynolds number that the critical velocity range may be expressed by the following two equations. For the lower critical velocity, below which the flow is always plastic, the equation is

$$V_{lc} = \frac{1000\eta + 1000\sqrt{\eta^2 + \dfrac{D^2 \tau_y \rho g}{3000}}}{D\rho} \ldots [3]$$

and for the upper critical velocity above which the flow is always turbulent the equation is

$$V_{uc} = \frac{1500\eta + 1500\sqrt{\eta^2 + \dfrac{D^2 \tau_y \rho g}{4500}}}{D\rho} \ldots [4]$$

The range between the lower critical velocity and the upper critical velocity represents a region of uncertainty as to whether plastic flow or turbulent flow will exist. Apparently the exact value of the critical velocity is controlled by the roughness of the pipe.

An equation for the critical velocity involving the roughness of the pipe is

$$V_c = \frac{8\eta + 8\sqrt{\eta^2 + \dfrac{f D^2 \tau_y \rho g}{24}}}{fD\rho} \ldots [5]$$

where f is the friction factor in Fanning's formula

$$H = \frac{2 fLV^2}{Dg} \ldots\ldots\ldots\ldots\ldots [6]$$

Solution of this equation must be done by successive approximations. The comparison between observed and computed values of the critical velocity shows close agreement when the roughness of the pipe is accurately known.

To check the validity of equations 3, 4 and 5, tests were made with various types of sludges to observe the critical velocities together with the values of the yield value, τ_y, and the rigidity, η. The results of these tests, together with observed and computed values of critical velocity taken from various tests reported in the literature, show a high degree of correlation between observed and computed values.

Turbulent Flow

It has been found mathematically and experimentally that for any pipe a plot of the friction factor, f, in equation 6, against the Reynolds number, as shown in Fig. 5, will show friction losses resulting from the flow of any liquid through any pipe or pipes, provided the roughnesses of the various pipes are dynamically similar. To determine the friction factor for any pipe by the use of such a chart the line to be used should represent the same relative roughness as that of the pipe to be used. Since the chart correlates all factors affecting flow it would be desirable to be able to plot the flow of all types of sludge on a similar diagram. This can be done provided a constant can be found for the sludge similar to the

---

Symbols

D = diameter of pipe, ft;
f = friction factor in equation 1;
g = acceleration due to gravity, ft per sec per sec;
H = difference in static head between two points in a pipe, ft of flowing substance;
L = length of pipe, ft;
(Re) = Reynolds number, $DV\rho/\mu$, dimensionless;
V = mean velocity of flow in pipe, ft per sec;
V_c = critical velocity, ft per sec;
V_{lc} = lower critical velocity, ft per sec $[(Re) = 2000]$;
V_{uc} = upper critical velocity, ft per sec $[(Re) = 3000]$;
μ = coefficient of viscosity, lb per sec ft;
η = coefficient of rigidity, lb per sec ft;
ρ = density of flowing substance, lb per cu ft;
τ_p = shearing stress in a flowing material at the boundary or pipe wall, lb per sq ft;
τ_y = shearing stress at yield point of plastic material, called yield value, lb per sq ft.

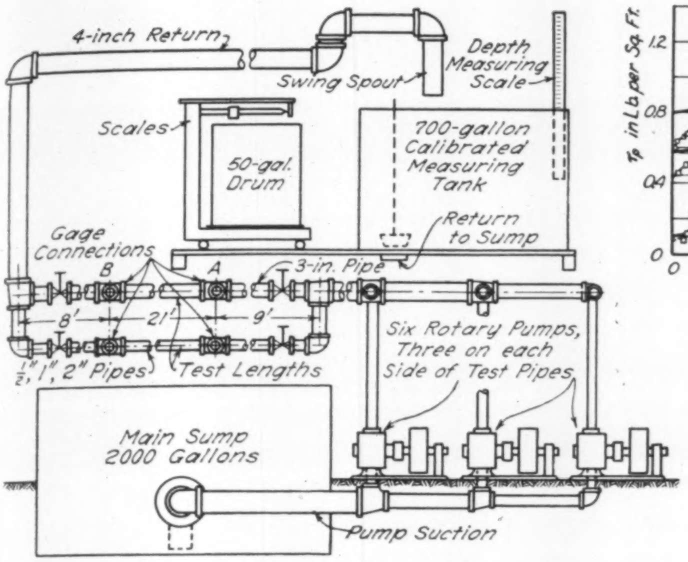


Fig. 3—Diagram of sludge flow apparatus

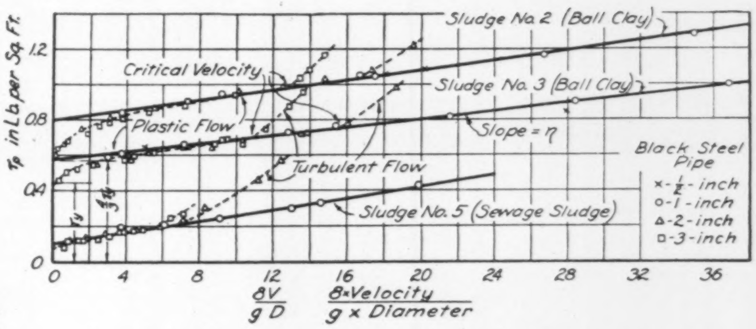


Fig. 4—Experimental verification of plastic flow equation

viscosity of a liquid. It has been found, in our investigation, from the results of about 900 tests using eight different sludges, that the viscosity of the dispersion medium of the sludges tested can be substituted for the viscosity of the liquid in the Reynolds number, and a friction factor chart constructed which will be approximately the same as the chart obtained for a liquid. Although the viscosities of the dispersion media in the sludges tested are closely equal to that of water, it is possible that similar results might be secured with dispersion media of higher viscosities.

The use of the viscosity of the dispersion medium in place of the viscosity of a true liquid in the Reynolds number is reasonable when it is recalled that in turbulent flow the frictional losses are due essentially to impact losses which depend only on density and velocity. The presence of suspended particles increases the density of the material, but in no way affects the basic viscosity of the dispersion medium. It is the viscosity of the dispersion medium which causes a relatively thin layer of material to move in laminar flow along the walls of the pipe while the material in the center of the pipe is flowing turbulently. In the flow of true liquids the thin layer at the pipe wall is called the "boundary layer." It is due to the boundary layer that the exponent of the velocity factor in the common hydraulic formulas is not two, as would be expected if the flow were wholly turbulent. It is found, however, that for very rough pipes or for very high velocities in smooth pipes the exponent does become two, indicating that flow is wholly turbulent and that frictional losses are due only to kinetic energy impact losses, the boundary layer having been reduced to zero thickness.

The most complete summary of available pipe flow data has been prepared by Emory Kemler and correlated on a friction factor diagram by R. J. S. Pigott. A reproduction of this friction factor chart, suitable for use in determining friction factors for most types of pipes, is given in Fig. 5. It has been found in our investigation, as shown in Fig. 7 and as explained above, that such a friction factor chart can be used for solving sludge flow, friction loss problems, provided the viscosity of the dispersion medium of the sludge is used in computing the Reynolds number.

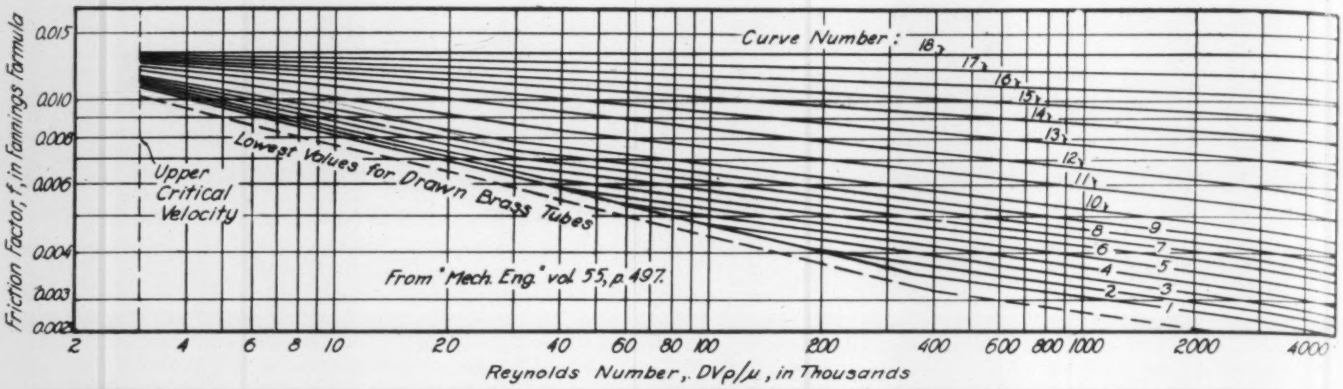
In the use of the chart, the Reynolds number is first computed using the viscosity of water as that of the sludge dispersion medium. Values of the viscosities of water at various temperatures are given in Table 3. Using this Reynolds num-

Table 1—Comparison of Observed and Computed Values of Head Loss in Pipes of Various Sizes

Type of Sludge and Source	Per Cent Moisture, by Weight	Diameter of Pipe, In.	Velocity, Ft per Sec	Observed Head Loss, Ft per 100 Ft	Computed Head Loss, Ft per 100 Ft	Per Cent Variation
Sewage sludge— Calumet Treatment Plant, Chicago $\tau_y=0.060$ $\eta=0.021$[a]	88	5	0.5	1.32	1.34	1.5
			1.0	1.42	1.43	0.7
			1.5	1.52	1.53	0.7
			2.0	1.62	1.63	0.6
			2.5	1.72	1.73	0.6
			3.5	1.92	1.92	0.0
			4.5[b]	2.18	2.17	—0.5
Sewage sludge— Calumet treatment plant, Chicago $\tau_y=0.014$ $\eta=0.035$[a]	90	12	0.5	0.144	0.150	4.2
			0.7	0.152	0.161	5.9
			0.9	0.162	0.176	8.6
			1.0	0.169	0.178	5.3
			1.5	0.200	0.214	7.0
			2.0	0.242	0.234	— 3.3
			2.6[b]	0.300	0.268	—10.6
Sewage sludge— Calumet treatment plant, Chicago $\tau_y=0.017$ $\eta=0.026$[a]	90	8	0.5	0.260	0.265	1.9
			0.7	0.275	0.287	4.4
			0.9	0.300	0.315	5.0
			1.5	0.372	0.361	— 3.0
			1.8[b]	0.480	0.408	—15.0
Clay slurry from water purification plant $\tau_y=0.011$ $\eta=0.005$[a]	86.4	4	0.5	0.30	0.32	6.7
			0.7	0.31	0.34	9.7
			0.9	0.33	0.35	6.1
			1.5	0.37	0.40	8.0
			1.8[b]	0.41	0.41	0.0
Sewage sludge— digested, sewage disposal plant, Stuttgart, Germany $\tau_y=0.010$ $\eta=0.077$[a]	90	7.9	1.0	1.50	1.57	4.7
			1.5	1.62	1.72	6.2
			2.0	1.82	1.86	2.2
			2.5	2.00	2.00	0.0
			3.0	2.20	2.14	— 2.7
			4.0	2.47	2.33	— 5.7
			5.0[b]	2.75	2.71	— 1.5
Illinois yellow clay suspension tests made in our investigation $\tau_y=0.72$ $\eta=0.028$[a]	52	3	2.0	26.1	26.3	0.8
			3.0	27.2	27.1	— 0.4
			4.0	27.9	28.0	0.4
			5.0	28.9	28.8	— 0.3
			6.0	29.7	29.7	0.0
			7.0[b]	30.5	30.5	0.0

[a] These values of τ_y and η have been determined by plotting appropriate data similarly to Fig. 2 and reading intercepts and slopes.

[b] Observed critical velocity.



Type of Conduit	Curve Number in Diagram Above																	
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Drawn Tubing-Brass, Tin, Lead, Glass, Diameter in Inches →	0.35 up													0.125				0.0625
Clean Steel, Wrought Iron, Diameter in Inches →	72	48-66	14-42	6-12	4-5	2-3	$1\frac{1}{2}$	$1-1\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$					
Clean Galvanized Iron Diameter in Inches →			30	10-24	6-8	3-5	$2\frac{1}{2}$	$1\frac{1}{2}-2$	$1\frac{1}{4}$	1	$\frac{3}{4}$	$\frac{1}{2}$		$\frac{3}{8}$				$\frac{1}{8}$
Best Cast Iron, Cement, Light Riveted Steel Ducts Diameter in Inches →			48-96	20-48	12-16	5-10	3-4	$2-2\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1							
Average Cast Iron, Rough Formed Concrete Diameter in Inches →			96	42-96	24-36	10-20	6-8	4-5	3									
First Class Brick, Heavy Riveted Steel Diameter in Inches →			220	84-204	48-72	20-42	16-18	10-14	8	5	4	3						

Note—In drawn tube, actual diameter is given. In pipe, nominal size of standard weight is given.

Fig. 5—Friction factor chart

ber and the curve representing the type of conduit to be used, the corresponding friction factor f, is found. The total head loss is then computed by means of equation 6.

It is to be noted that for high friction factors a relatively large error in determining the viscosity of the dispersion medium will have but slight effect on the value of the friction factor, and that the magnitude of this effect is less than that resulting from a slight error in the assumption of the pipe wall roughness.

In order to determine the roughness of the test pipes in our apparatus, runs were made with water in which the velocity of flow, head loss, density and temperature were observed. Results of these runs are shown on the friction factor—Reynolds number chart of Fig. 6. The friction factors for the 1 in. and 2 in. pipes are on the line predicted for the commercial black steel pipe. The 3 in. pipe has slightly smaller friction factors than the 1 in. and 2 in. pipes, indicating a smoother interior. The ½ in. pipe was noticeably rougher and the friction factors were found to be correspondingly higher.

Fig. 7 is the friction factor chart correlating all data on the turbulent

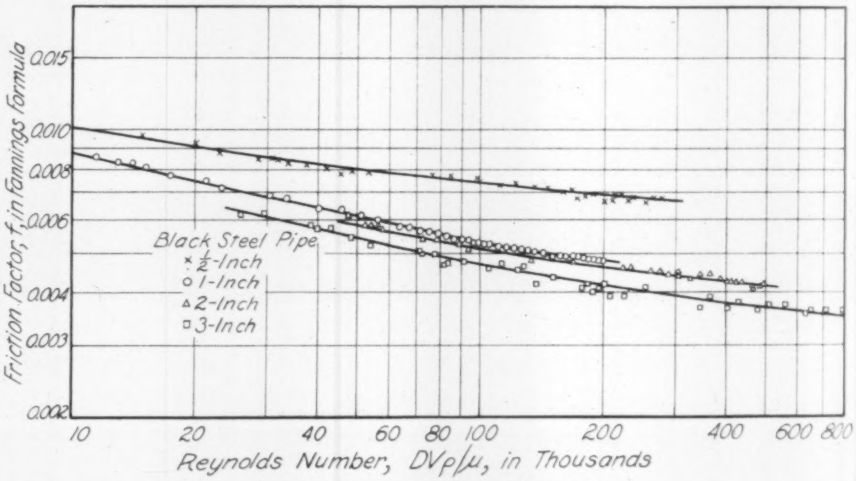


Fig. 6—Friction factors for test pipes

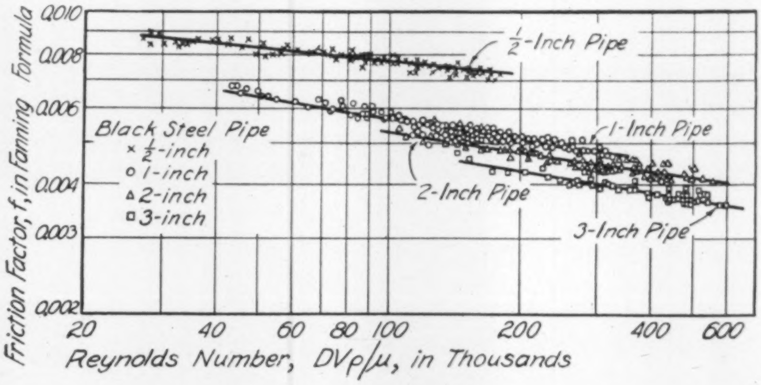


Fig. 7—Results of turbulent flow tests

flow of sludge taken in our investigation. On comparing this figure with Fig. 6, which was obtained from the same pipes using a true fluid (water), it is seen that the agreement between the two figures is good. Friction factors obtained from the sludges are, as a whole, slightly higher than those obtained for water. This fact might have been caused by some corrosion occurring in the pipes during the tests, which extended over a six month period. The agreement, however, is within 5 per cent, which is within the degree of precision of the determination of the roughness of the pipe.

From the results of these tests it has therefore been assumed that the ordinary friction factor chart may be used for computing sludge flow friction losses in pipes *as long as the flow is turbulent.* To determine whether the flow is turbulent the critical velocity should be computed by means of equation 5.

Table 2—Comparison of Computed and Observed Values of Critical Velocities

Description of Sludge	Diameter of Pipe, In.	Yield Value τ_y	Coefficient of Rigidity η	Critical Velocity			Source of Information
				Computed		Observed	
				V_{lc}	V_{uc}		
Imhoff Tank Sewage Sludge	8	0.057	0.044	4.2	5.7	4.0	Reference No. 4
	8	0.043	0.040	3.8	5.1	3.7	
	8	0.033	0.036	3.3	4.5	3.5	
	8	0.024	0.034	2.9	3.8	3.0	
	8	0.017	0.026	2.4	3.2	2.7	
	8	0.013	0.018	1.9	2.6	2.2	
	8	0.006	0.016	1.5	2.0	1.8	
	8	0.003	0.016	1.1	1.6	1.3	
	8	0.0015	0.014	0.9	1.3	1.0	
	12	0.020	0.0514	2.8	3.8	3.0	
	12	0.014	0.035	2.1	3.0	2.6	
	12	0.009	0.032	1.8	2.5	2.3	
	12	0.005	0.036	1.6	2.2	2.1	
	12	0.0026	0.025	1.1	1.6	1.7	
	12	0.0015	0.024	1.0	1.4	1.4	
	3	0.115	0.0163	5.4	7.3	6.5	Reference No. 1
	2	0.091	0.0165	5.6	7.7	7.0	
	1	0.075	0.0165	7.6	11.1	7.0	
	1	0.065	0.0165	7.4	10.8	7.0	
Clay Slurry	4	0.020	0.005	2.0	2.7	2.2	Reference No. 3
	4	0.056	0.005	3.3	4.2	3.6	
	4	0.117	0.005	4.6	5.8	6.0	
	4	0.158	0.005	5.3	6.8	7.0	
	4	0.214	0.005	6.1	7.6	8.5	
	4	0.012	0.005	1.5	2.1	1.8	
				Friction Factor	V_c		
Clay Slurry	½	0.60	0.015	0.0079	14.0	14.2	Reference No. 2
	1	0.60	0.015	0.0055	14.9	14.7	
	2	0.60	0.015	0.0051	13.5	13.5	
	3	0.60	0.015	0.0045	13.8	13.8	

Table 3—Viscosity of Water at Various Temperatures
(*Bingham and Jackson, National Bureau of Standards Bulletin* 14, 75, 1918)

Temperature, F	Viscosity, Lb per Ft Sec	Temperature, F	Viscosity, Lb per Ft Sec
32	12.1×10^{-4}	80	5.8×10^{-4}
35	11.4×10^{-4}	85	5.4×10^{-4}
40	10.4×10^{-4}	90	5.1×10^{-4}
45	9.5×10^{-4}	95	4.9×10^{-4}
50	8.8×10^{-4}	100	4.6×10^{-4}
55	8.1×10^{-4}	105	4.4×10^{-4}
60	7.6×10^{-4}	110	4.2×10^{-4}
65	7.0×10^{-4}	115	4.0×10^{-4}
70	6.6×10^{-4}	120	3.8×10^{-4}
75	6.2×10^{-4}	125	3.6×10^{-4}

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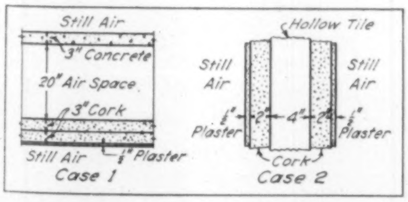
How Can I Figure the Heat Transfer Coefficients for These Walls?

A subscriber to HPAC asks for advice on calculating the heat transmission coefficients for the two wall constructions roughly illustrated in the sketch. The answers have been prepared by Paul D. Close, technical secretary of the Insulation Board Institute.



Case 1—The air space conductance of 1.10 Btu per hr per sq ft per F is for thickness stated or used on construction, not per 1 in. Therefore, this value would be used for the entire 20 in. air space rather than on a per inch basis and the coefficient of transmission *(U)* for the construction involved, for still air on both sides, using factors from the ASHVE Guide, would be:

$$\frac{1}{\frac{1}{1.65}+\frac{3}{12.0}+\frac{1}{1.10}+\frac{6}{0.30}+\frac{0.5}{3.3}+\frac{1}{1.65}}$$

$$=\frac{1}{22.53}=0.044 \text{ Btu per hr per sq ft per F temperature difference.}$$

This, of course, is not a type of construction that would generally be used in practice as there is no advantage from the heat resistance standpoint of a 20 in. air space over a 1 in. air space. To be strictly correct, different air space conductances should be used for upward and downward heat flow through horizontal surfaces.

Case 2—The conductance of 4 in. hollow tile as given in the ASHVE Guide is 1.00. The resistance of the 4 in. of hollow tile would, of course, be 1/1.00. The coefficient *U* is calculated as follows:

$$\frac{1}{\frac{1}{1.65}+\frac{0.5}{3.3}+\frac{2}{0.3}+\frac{1}{1.00}+\frac{2}{0.3}+\frac{0.5}{3.3}+\frac{1}{1.65}}$$

$$=\frac{1}{15.85}=0.063.$$

The once-through assembly line method is used in this paint shop. Note the wire filters in the ceiling through which the air is drawn downward through the grille work in the floor

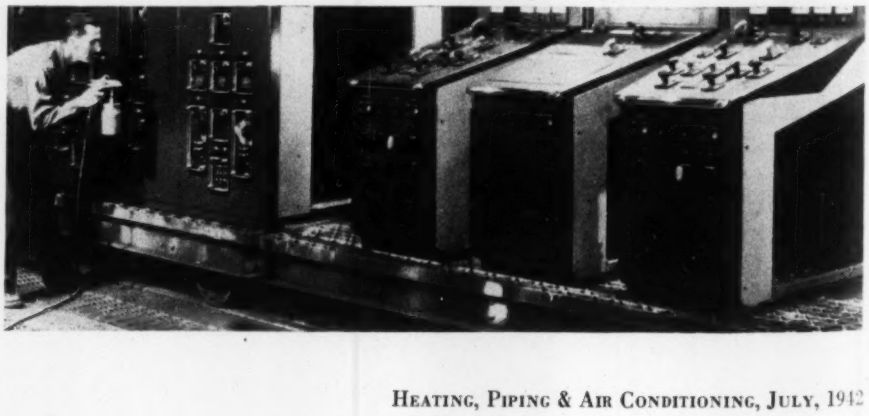
AIR FILTERS COVER ENTIRE CEILING OF PRODUCTION PAINT SHOP

STRAIGHT LINE production methods are used at the recently completed paint shop at one of the Westinghouse Electric works. Switchgear panels and switchboards are brought in for painting on flat top trucks that run on tracks from the assembly aisle through the paint shop and on out to the shipping floor.

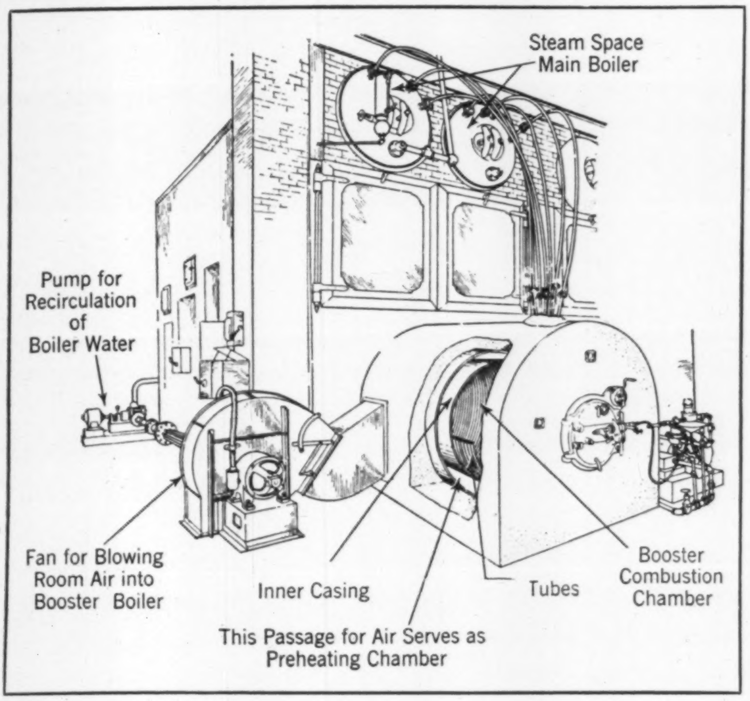
A special feature of the 70 ft by 25 ft by 15 ft booth is the method of handling the clean ventilating air. A downdraft system is used whereby air from outside the booth is drawn through the ceiling. The air is cleaned as it passes through 180 wire mesh washable filters, each 20 by 20 by 2 in. thick, covering the entire ceiling area. Paint spray in the room is carried downward with the air through open grille work in the floor and is deposited on a water surface in a pit covering nearly the area of the floor. The water in the pit is recirculated by four pumps. Make up water is supplied to retain the proper water level.

The air drawn downward through the floor grille is expelled to the shop by four fans, each driven by a 15 hp motor. Total air exhausted is about 200,000 cfm.

Finishing touches are given to complicated switchboards and switchgear panels in a new production paint shop. The units are moved in from the assembly aisle on small flat cars.



Booster Triples Capacity of 36 Year Old Boiler


Steam Space
Main Boiler
Pump for Recirculation of Boiler Water
Fan for Blowing Room Air into Booster Boiler
Inner Casing
This Passage for Air Serves as Preheating Chamber
Tubes
Booster Combustion Chamber

SUMMARY — A forced recirculation booster unit, little larger than a domestic furnace, jumps the capacity of a boiler at the New York department store of B. Altman and Co. from 250 to 850 hp. Details of the installation are shown in the sketch. The exposed pipes leading to the steam space of the main boiler, ordinarily insulated and of slightly different construction, are exposed here for testing purposes. . . . This article is reproduced by permission from the Inco Magazine, issued by the International Nickel Co.

What seems in these days of excessive demands on power plants the answer to a power engineer's prayer is a booster unit that has been found to perform some near miracles. It has been found, in one case at least, to increase the power output of a 36 yr old boiler from 250 to 850 hp.

The unit, designed by the late W. D. LaMont, a former naval officer, originally was intended for marine purposes. To date its chief applications, however, have been on land, one of them in the New York department store of B. Altman and Co.

Boiler water is drawn from the drum of the boiler at a constant rate of 85,000 lb per hr and pumped direct into tube coils in the wall of the combustion chamber of the booster. All combustion takes place in this chamber. Hot exhaust gases from this chamber pass into what once was the combustion chamber of the old boiler at a temperature of approximately 1800 F. Thus the main boiler becomes what actually is a waste heat unit.

The recirculated water from the boiler is heated in the coiled boiler tubes of the booster where about 25 per cent is formed into steam. This steam, with entrained water, is then passed into the steam space of the main boiler. From this point the hot exhaust from the booster, forced by fan into the boiler, takes over the task of transforming the water in the boiler into steam for the power plant as a supplement to the steam from the booster. Exhausted steam is condensed and passed back for another recirculating cycle.

The fan forces room air between an inner and outer casing surrounding the booster's combustion chamber. This preheats the air as it flows into the chamber.

The value of the boiler booster lies in the speed with which it transfers heat to the water during the few seconds that the water flows through the tube coil in the firing chamber. Since the naked tube coil must be exposed directly to intense heat, only a material with high strength and exceptional resistance to elevated temperatures can be used. Furthermore, since even a thin coating of scale or oxide, or corrosion products seriously slows down the rate of heat transfer, it is important, therefore, that the tube material resist corrosion by fuel gases, heat, and boiler water and retain its clean surface and fast heat transfer rate. High nickel-chromium alloy tubes were found to be the answer. Ability of the high nickel-chromium alloy to retain its high physical properties at high temperatures makes it feasible to design tube sizes with minimum wall thickness for fast heat transfer and economy of metal with adequate safety. This alloy also is used for the casing, heated to a temperature of 1000 F.

AIR CONDITIONING SPEEDS TORPEDO BOAT PRODUCTION

Production of P-T torpedo boats is aided at a Southern plant by a 60 ton air conditioning system which cools the administrative, drafting, training, and personnel offices.

On the top floor of the three-story building is the huge drafting department which lays out plans for each boat; on the second floor, the general offices and management departments, purchasing and naval offices; and on the first floor, a publicity department, personnel office, and a group of classrooms for training pilots, navigators, and crews. In each of these departments, air conditioning was essential because of 100 deg summer temperatures and 90 per cent humidity conditions. Without air conditioning, according to the management, loss of man hours through illness, mistakes in drafting, etc., would retard production of torpedo boats. Particularly in the drafting room on the top floor, exposed to a heavy sun load magnified by a metal roof, was it necessary to provide efficient conditions for draftsmen working on blueprints and specifications.

**GET IN THE SCRAP!
SALVAGE
FOR VICTORY**

Ventilation of Rotor Casting Department Given Special Attention by Motor Maker

Volume of motor production has grown to such an extent at the Reliance Electric & Engineering Co. that several additions to the manufacturing facilities have been required. Latest to be substantially enlarged is the rotor casting department, which has been transferred to a building of its own.

In the casting operation molten aluminum at 1250 to 1420 F is poured into the press, the mold is moved into position, the safety hood of the press closed and the hot metal forced up into the mold. One press is air operated, the other hydraulically operated. The casting operation takes only a few seconds, but pressure is maintained on the mold while the fumes are expelled from the interior of the press by a blower and the metal allowed to chill.

The problem of securing good ventilation for the department was given special attention and provision made for changing the air as frequently as three times a minute, if desired.

A 6 ft by 6 ft ventilating hood, equipped with a 10,000 cfm exhaust fan, was installed directly above the two casting presses. Special vent pipes in which are installed motor driven blowers force the fumes from the presses up into this exhaust hood, which in turn discharges them through a stack.

Two exhaust fans mounted along the wall on either side of the hot metal ladles and driven by 1½ hp motors, are used to remove the excess heat during the warm summer months when the building doors are open. They discharge through shuttered apertures in the steel sash of the building's side wall.

Fan enclosures, shelves for the motors and supporting frames were made from sections of steel plate and welded together. The vertical channels between which the fans are mounted are anchored at floor and ceiling, but kept separate from the walls and steel sash to reduce noise and vibration.

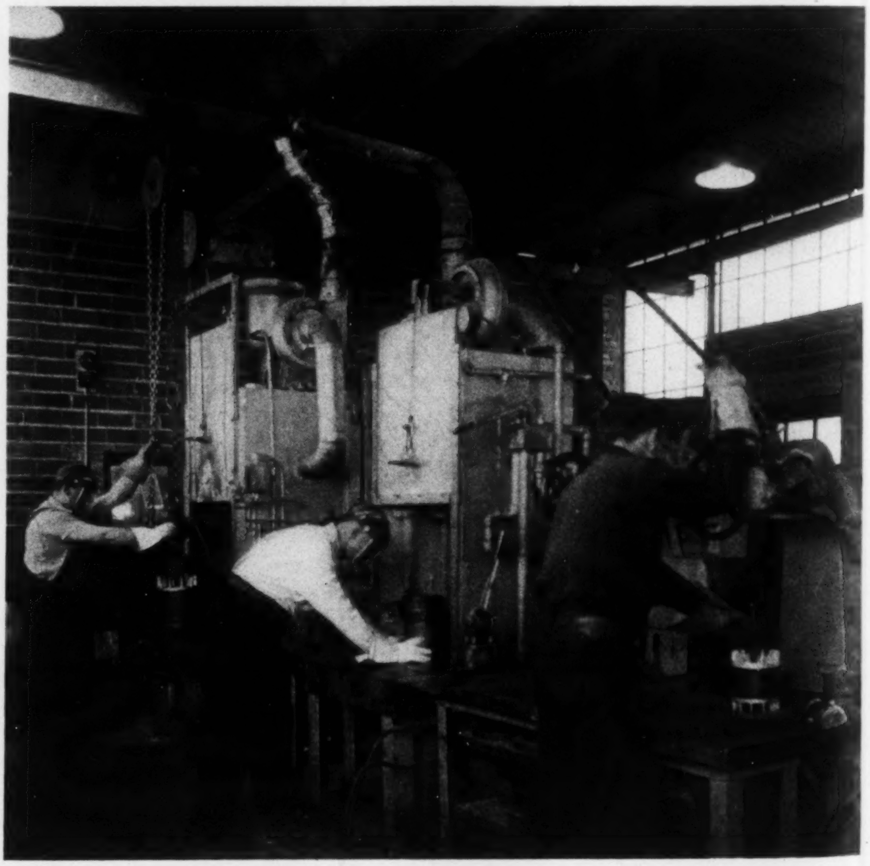
A separate vent line and motor driven exhaust blower, discharging directly to the outside of the building, serve the preheating oven.

Some of the molds and dies used are rather heavy; there are two overhead chain pull hoists to handle them. These hoists travel the length of the casting room, each providing 100 per cent coverage of half of the area.

To permit the hoist serving the casting presses to pass back and forth freely and without interfering with the operation of the ventilating system for these units, the supporting cross member of the hoist was placed so that it slips between the discharge ends of the vent lines from the presses and the opening of the ceiling exhaust hood. In operation it has been found that the suction created by the 10,000 cfm fan in this hood is sufficient to carry the discharge from the vent lines across the intervening 6 in. between them and the entrance to the ceiling hood.

The enclosure of the two aluminum melting pots was designed with particular attention to obtaining efficient and safe operation, and can be readily removed to give access to the furnaces. Two inch insulation was used in the sides of the enclosure and 4 in. of insulation in the tops. These tops were made in two sections to permit only limited exposure of the hot metal for such operations as adding ingots, skimming off the slag, inserting the pyrometer rods, etc. To minimize further the loss of heat the covers were equipped with spring seals and are motor operated to open and close with a positive action. •

SUMMARY—The rotor casting department of the Reliance Electric & Engineering Co. has been enlarged and moved to a building of its own. Special attention was given to ventilation. There is a 6 ft by 6 ft ventilating hood equipped with a 10,000 cfm exhaust fan above the two casting presses. To permit the hoist serving the casting presses to pass back and forth freely without interfering with the ventilating system, the supporting cross member of the hoist was placed so that it is between the discharge ends of the vent lines from the presses and the opening of the exhaust hood, as shown in the picture. It has been found that the suction created by the fan is sufficient to carry the discharge from the vent lines across the intervening 6 in. between them and the exhaust hood



Heating·Piping and Air Conditioning

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

Journal *Section*

Only 90 Days To Prepare

On June 25 the Society's War Service Committee delivered its findings and recommendations to Secretary of the Interior Harold L. Ickes, the Coal and Petroleum Coordinator.

In response to the offer of the Society to make its research and other facilities available to the government, Secretary Ickes requested the Society's cooperation in finding a solution for the current fuel supply problem which faces the eastern states and the Pacific Northwest during the coming winter.

The members of the War Service Committee who presented the report, Chairman B. M. Woods and W. L. Fleisher, made it plain that current surveys must accurately determine the amount of fuel (coal and oil) that will be available in the critical areas and offered four specific methods, which could be progressively applied to meet even the most drastic fuel conservation program, and suggested specific methods to inform the public promptly concerning economic and speedy methods of adapting or installing accessory equipment to meet the existing emergency.

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Every member can render a vital public service NOW by campaigning for Fuel Savings by every possible method.

TIME IS SHORT—THE HEATING SEASON STARTS IN 90 DAYS

CONTENTS

JULY, 1942

Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson 432

Overloading of Viscous Air Filters During Accelerated Tests, by Frank B. Rowley and Richard C. Jordan 438

Graphical Symbols for Drawings 448

ASHVE Members on Active Duty 453

Local Chapter Reports 454

Candidates for Membership 460

Candidates Elected 460

Officers, Council and Committees 461

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Trend Curves for Estimating Performance of Panel Heating Systems

By B. F. Raber* and F. W. Hutchinson,** Berkeley, Calif.

SUMMARY—From an analysis of trend curves established for a particular structure by the application of rational equations previously developed, it is shown that the ceiling is the most effective location for radiant heating panels. The performance of such panels becomes more effective as outside temperatures decrease and panel surface temperature, ventilation rate and structure conductance increases

THE PERFORMANCE of a panel heating system can be determined with reasonable accuracy by carrying out a detailed analysis based on rational principles.[1] Likewise, a correlation of experimental data from other installations would be expected to afford a background from which one might estimate the performance of a new system. Unfortunately, however, each of these methods is subject to severe objections: The rational analysis is difficult, laborious and time-consuming while the experimental correlation is limited through lack of satisfactory data from a great enough number of different installations and is further handicapped by the technical problem of extrapolating such data to the conditions of a new design.

Basically, the increased complexity of calculations for the panel system over similar calculations for other types of heating systems arises almost entirely from the fact that, with panel heating, selection of a fixed design inside air temperature is not admissible nor possible. As the MRT (*mean radiant temperature*) increases, the corresponding inside air temperature for a constant sensation of warmth must decrease. The MRT is influenced by the following factors: panel surface temperature, outside air temperature, location of heating surface, ventilation rate and heat transmission characteristics of the structure. Thus any change in one or more of the factors outlined will result in a change of MRT and a consequent variation in the value of the inside air temperature corresponding to optimum comfort.

The essential point of difference between panel heating and other methods of heating is the reduction of the inside air temperature which must occur (for comfort) when the fraction of energy transferred by radiation is increased. As the radiant fraction becomes greater the MRT of the enclosure must increase and the inside air temperature decrease; as the convective fraction increases the opposite sequence must occur. Such considerations suggested that an acceptable criterion of effectiveness for radiant heating systems should indicate the difference between the MRT and the inside air temperature. Such a criterion is defined by the equation,

$$D = \text{MRT} - t_a \ldots\ldots\ldots\ldots\ldots (1)$$

where D is the depression of inside air temperature, t_a, below the mean radiant temperature.

A knowledge of the magnitude and manner of variation of D is a first requisite in the design or performance analysis of a proposed or existing panel heating system. If the depression is large, the heating load on the panel heated structure may differ greatly from the equivalent heating load for a convective system; conversely, if D is small the load will be approximately the same for either panel or convective heating and the need for precise panel analysis will be obviated.

With respect to operating performance, D is significant for two reasons:

1. Systems for which D is small throughout the operating range can be thermostatically controlled at constant air temperature in exactly the same way as convective heating systems. When D varies widely with operating conditions, the optimum inside air temperature must likewise vary, and simple thermostatic control based on constant air temperature cannot be used effectively. In the past, ill-advised criticism of radiant heating has resulted from systems having large variation in D, yet controlled to operate at a constant air temperature. In such cases room conditions inevitably will be too warm for all operating loads greater than the one corresponding to the thermostat setting and too cool for all operating loads less than that corresponding to the particular condition for which the selected air temperature does correspond to optimum comfort.

2. Two of the most important advantages claimed for radiant heating are the greater feeling of freshness and the reduced ventilation heating load which accompany operation at reduced air temperature. Thus changes in D are indicative of the varying effectiveness of panels, under different design or operating conditions, in realizing the advantages which are primarily responsible for their use. Any change in design or application which serves to reduce D likewise serves to reduce the effectiveness of the panels in transferring a maximum fraction of energy by radiation, and hence causes the panel system to approach more closely the performance of a conventional convective system. Hence, D varies directly (but not proportionately) with panel effectiveness; the greater D becomes, the more closely does the panel system approach the performance of an ideal radiant heating system.

Trend Curves

Based on D as an acceptable criterion of panel performance, the intent of this paper is to present a series of simplified general trend curves to assist the designer in estimating the effect on performance of changes in any one or more of the five principal design factors. Numerically, the magnitude of D and

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[1] Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Paul, Minn., June, 1942.

its rate of change with any one variable depend on the specific conditions of the particular structure for which the design or analysis is intended. For estimating purposes, however, numerical values are frequently unnecessary, but information is needed as to the general effect on performance of a proposed change in some one factor.

The trend curves here presented are numerically significant only for the particular structure used in the calculations, but the slope of each curve and the per cent change in D with changes in each of the other factors can be taken as representative for any type of structure when operating at any point within the limits of variation which are used in these curves. Judicious examination of the rate of change of D with each variable makes possible the use of the curves as a rough basis for estimating the numerical magnitude of D for any other structure.

Specifically, the trend curves have been constructed to provide direct and universally valid answers to five questions that recur constantly during the design or analysis of any panel heating system. These are:

1. For a given panel installation, in what way does operating effectiveness vary with the heating load (or, specifically, with the outside air temperature, t_o)?
2. For a given system operating at design outside temperature, how does effectiveness vary with the ventilation rate?
3. For a given structure with fixed ventilation rate and constant outside temperature, how does effectiveness vary with design panel surface temperature (or, conversely, how does effectiveness vary with design panel area)?
4. For fixed ventilation rate, constant outside temperature, and fixed panel area, how does effectiveness vary with the overall coefficient of heat transmission of the structure?
5. For a given structure with fixed ventilation rate, constant panel area and fixed outside temperature, how does effectiveness vary with the location of the panel (as floor, wall, or ceiling)?

Answers to these five questions are presented in Figs. 1 to 5, respectively, each of which will be discussed in a later section.

Construction of Curves

The calculations leading to the trend curves are based entirely on equations taken from an earlier contribution of the authors.[2] For this reason derivations are not included here, but are briefly referred to in the appendix. Need exists, however, for an explicit statement of the values of certain basic constants and fundamental empirical relationships which have been used in the calculations. These are as follows:

1. Film coefficients of heat transfer by *convection only* (Based largely on ASHVE Report No. 1098).

   a. For ceiling panels = 0.40 Btu per hr per sq ft per deg F.
   b. For wall panels = 0.70 Btu per hr per sq ft per deg F.
   c. For floor panels = 1.10 Btu per hr per sq ft per deg F.
   d. Equivalent value for all unheated room surfaces = 0.80 Btu per hr per sq ft per deg F.

2. Equivalent film coefficient for transfer by *radiation only*:[3]

   = 1.00 Btu per hr per sq ft per deg F

3. Comfort is considered as being realized in a room with both air and walls at 72 F. The comfort curve for radiant heating is assumed to pass through the point $t_a = 72$ when MRT $= 72$ and to have slope of unity. Any error which may exist because of this assumption would result in a change of the position, but not the trend of the curves for D.

4. The design outside temperature has been taken as $t_o = 0$ F.

5. The design panel surface temperature has been taken as $t_p = 120$ F.

6. The design ventilation rate corresponds to an air flow of 2.5 cu ft per hour per square foot of inside surface. This is equivalent to one air change per hour in a room 20 ft x 20 ft x 10 ft.

7. The design over-all coefficient of heat transfer is assumed constant for all unheated surfaces, and is taken as numerically equal to 0.250 Btu per hr per sq ft per deg F. This is equivalent to a conductance from the inside surface to the outside air of 0.29 Btu per hr per sq ft per deg F. To simplify the calculations by eliminating the necessity of considering the shape factors of one wall with respect to others, the inside surface temperature has been taken as uniform over all unheated surfaces. Calculation from the precise equations indicates that each of these assumptions leads to a conservative value of D. As a first approximation, therefore, one can neglect shape factors and treat any real structure as though it were constructed uniformly of a material having a conductance (calculated from inside surface to outside air) equal to the average weighted conductance of the real structure. In such cases the conductance of inside partitions or floors having no transmission losses would obviously be taken as 0.

The assumptions of relationship 7 obviate the need for considering the effect of shape factors in applying the heat balance equations for performance analysis, but it is important to note that nothing is said as to the uniformity of heating effect *at the occupant*. This paper deals only with the over-all sensation of warmth of an occupant in the room; the comfort of that occupant will also depend on the uniformity of irradiation received at the body surface. This, in turn, depends on proper distribution of panel surface as dictated by analysis from consideration of shape factors of the panels with respect to the occupant.

Effect of Load on Performance

The variation of D with load is shown graphically in Fig. 1. Having selected an over-all coefficient of 0.250 and a ventilation rate of 1 air change per hour the necessary area of panel (at 120 F surface temperature) was calculated to provide optimum comfort when the outside temperature is 0 F. A panel area equal to 22.5 per cent of the total inside surface (90 per cent of the ceiling area in a room 20 ft x 20 ft x 10 ft) was found to be required and it was then assumed that this entire area would remain in operation at all higher outside temperatures. With constant panel area the panel surface temperature must obviously vary from a maximum of 120 F at

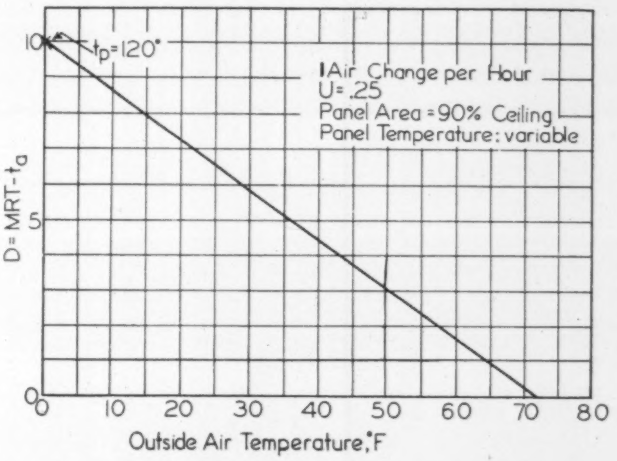

Fig. 1—Panel effectiveness *vs.* outside temperature

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[2] Loc. Cit., Note 1.
[3] Loc. Cit., Note 1.

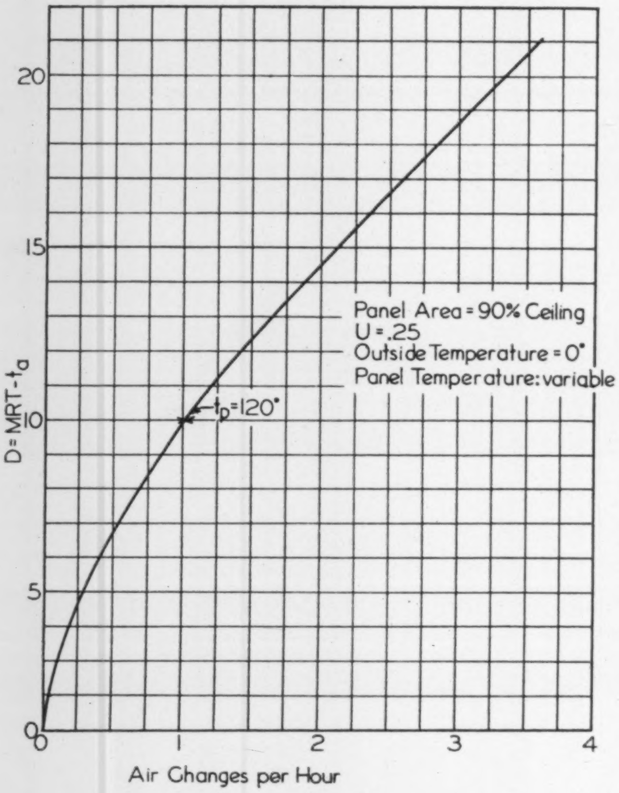

Fig. 2—Panel effectiveness *vs.* ventilation rate

0 F outside, to 72 F at 72 F outside. Analysis showed the variation of D with t_o to be a straight line with maximum $D = 10$ F when greatest load occurs and D reaching 0 as the heating load approaches 0.

The curve of Fig. 1 is significant for the following reasons:

1. Since D increases with load, the effectiveness of any panel system in permitting operation with reduced inside air temperature is greatest when the outside temperature is a minimum. This means that the saving on ventilation air heating becomes progressively more significant as the average load increases. Thus the advantage of reduced ventilation load as an argument for panel heating takes on greater significance in localities where the heating season is extended and the load severe than in climates where operation is usually at low load.

2. From the standpoint of health and comfort the desirability of reduced inside air temperature increases as the outside temperature drops. As shown in Fig. 1, a panel system provides the greatest reduction at the time when that reduction is most welcome. With very low outside temperatures, reduced inside air temperatures (with optimum comfort) alleviate to some extent the low humidity condition found in heated rooms, reduce the feeling of stuffiness associated with entrance into a heated space from the outside, and possibly serve to reduce the shock effects associated both with entrance into and exit from a heated room. This latter point is subject to debate, since the heat balance at the body surface is the same regardless of air temperature, but, beyond doubt, the change in conditions in the mucous membranes of the nose and throat which are acted on by the room air, but not affected by the MRT, would be less severe in a panel heated room.

3. An additional factor of interest evident from this Fig. 1 is that tests of panel installations should always be conducted when the outside temperature is low. One determination of D at a low outside temperature will permit drawing accurately the D *vs.* t_o line for the entire operating range, but if the test is conducted at a high outside temperature, the value of D will be small, the numerical value of the experimental error unchanged, and the resultant error from extrapolating the D curve to lower outside temperatures may be very great. Because the variation of D with load is so large, test results from different installations cannot be compared correctly unless the data are first adjusted for differences in outside temperature at the time of the tests.

4. The conditions for which Fig. 1 was constructed are reasonably representative of average practice. Since the value of D at design outside conditions is 10 F, evidently the analysis of the panel system for such a structure based on principles of standard convective heating design might lead to serious error. In the case of this particular structure, however, the convective analysis would be conservative since calculation shows that the rational analysis permits a 5 per cent reduction in panel area over that which would be required if calculations were based on usual convective heating load determinations.

Effect of Ventilation Rate on Performance

Fig. 2 brings out the very significant fact that the ventilation rate is the most important single factor in determining the performance of a panel heating system. In a room with no mechanical ventilation and 0 infiltration there can be no difference between the MRT and inside air temperature. Obviously, therefore, data from tests on panel heating installations are valueless unless information is presented concerning the ventilation rate at the time of the tests. Regardless of construction, load, outside temperature or panel area, the value of D must approach 0 as the ventilation rate approaches 0. The only exception to this fact is that changes in the film coefficient for convection, with respect to the position of the surface, do influence the heat balance somewhat, and this effect would result in a small D, even at 0 air change. However, this factor is hardly of practical significance and does not alter the general conclusion that ventilation rate is the most important single factor which influences panel heating effectiveness.

The validity of the conclusion that D must be 0 (neglecting effects of changing film coefficients) when there is no ventilation, can be visualized readily in terms of equilibrium conditions in a sealed room. In such an enclosure heat would flow by convection from the panel to the air and from the air to the unheated surfaces; the air temperature would reach an equilibrium value at some point between panel and wall surface temperatures. Suppose that this equilibrium air temperature were less than the MRT. By the concept of MRT as an average or equivalent uniform surface temperature, one could set up an equivalent room with all surfaces

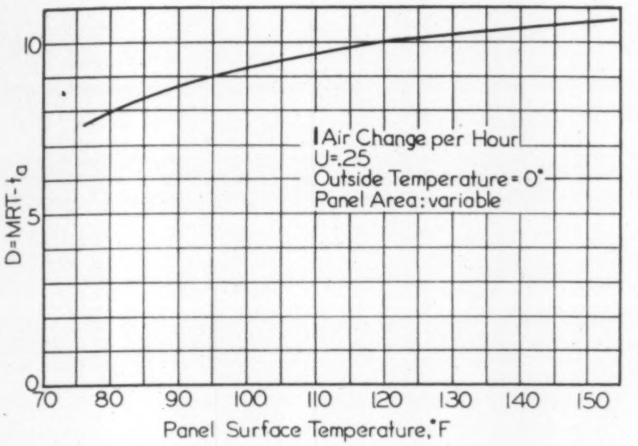

Fig. 3—Panel effectiveness *vs.* surface temperature

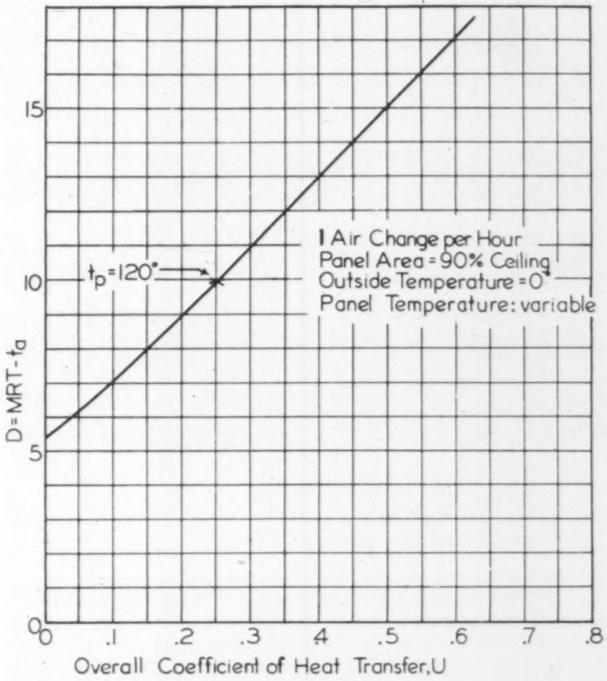

Fig. 4—Panel effectiveness *vs.* type of building construction

at a temperature equal to MRT, but if t_a were less than MRT there would be a flow of heat from the walls to the inside air which then could not be in equilibrium; it follows, therefore, that there cannot be a temperature difference between MRT and t_a.

The conditions for which Fig. 2 was plotted are similar to those for Fig. 1 and the D, 10 F, corresponding to 1 air change per hour, checks Fig. 1. Since the panel area is assumed constant, the panel surface temperature must vary as a function of ventilation rate; for more than 1 air change per hour the panel temperature will exceed 120 F while for lower ventilation rates the temperature will be less than 120 F.

The increase of D with ventilation rate is seen to be very rapid, the D corresponding to 3½ air changes per hour (specifically, 8.75 cu ft of ventilation air per hour per square foot of inside surface in a room 20 ft x 20 ft x 10 ft) being in excess of 20 F and the air temperature for optimum comfort under such conditions being therefore less than 62 F. The value of a large D as a means of reducing the ventilation air heating load is obviously most important in structures where the number of air changes is high; the figure also makes evident that the per cent saving on ventilation load through use of panel heating increases as the ventilation load becomes greater. However, even at ventilation rates corresponding to residential infiltration (½ to 1½ air changes per hour) the magnitude of D remains great enough to be of importance in designing and controlling a panel heating system.

Effect of Design Panel Surface Temperature on Performance

The question has frequently arisen as to whether low temperature panels of large area are more effective than small area, high temperature panels. Fig. 3 was constructed to show the effect on D of varying the design panel surface temperature when all other factors remain the same. Calculations for this figure were based on a constant value of the film transfer coefficient by convection from the panel surface; this condition was considered justified in that consideration of the variation of this factor would have slightly *reduced* the slope of the D *vs.* t_p curve and therefore have indicated an even smaller importance of panel temperature than is now evident from Fig. 3.

Referring to this figure, panel surface temperatures were investigated from 77 F up to 205 F; the value of D at 205 F was found to be less than 11 F, indicating that beyond 150 F panel temperature has practically no effect on the air temperature depression. Even in the range from 90 F to 160 F, corresponding to the extreme limits of present practice, the change in D is too small to be of real significance. The conclusion is unavoidable that heated surface temperature has very little effect on the operating performance of a panel system.

In connection with the temperature range of Fig. 3 one fact requires comment. The area of a panel operating at 80 or 90 F would exceed the ceiling area of the 20 ft x 20 ft x 10 ft room. This would mean that part of such a panel surface would operate as a wall rather than a ceiling unit. To permit direct comparison of data from all five figures this discontinuity effect was ignored and it was assumed that the installation would be in a room of such shape that the requisite panel area could be installed entirely in the ceiling. The fact is important, however, because it indicates existence of a lower limit to panel temperature in a given room (with given design outside conditions) beyond which the designer cannot go without using a split ceiling-wall panel installation.

Effect of Building Construction on Performance

Fig. 4 is based on the same general conditions as Fig. 1, that is, a panel operating at temperature 120 F establishes comfort in a structure for which the uniform over-all coefficient of heat transfer is 0.250 and the ventilation rate is 1 air change per hour. Outside temperature is constant at 0 F and the D for this set of conditions is = 10.0 F. Considering the panel area as fixed and allowing the surface temperature to change as a function of the over-all coefficient of heat transfer a relationship is established between *D* and the type of construction.

As insulation is added *D* decreases until for the perfectly insulated structure it assumes a minimum value based on 0 transmission losses and a heat balance between the inside surface and the ventilation air passing through the room. Note that the minimum value of *D* is determined by the ventilation rate, the outside temperature and the panel surface temperature. From previous curves one would expect this minimum *D* to become greater with increase in panel temperature or ventilation rate and with decrease in outside temperature.

The conclusion is evident that the effectiveness of panel installations is greatest in poorly insulated structures. This view is somewhat at variance with accepted opinion since it suggests that residences may present greater advantages for panel heating than do larger structures. Recent experience in the United States seems to substantiate this conclusion, but insufficient field data are yet available to permit experimental investigation of the analytical results.

Effect of Panel Location on Performance

The selection of floor, wall, or ceiling panels for a given installation is influenced by many economic and construction factors which may outweigh in importance the consideration of performance. When, however, performance is the controlling feature there is every reason to believe that ceiling panels are, by far, preferable to any other type. Fig. 5 considers the same structure as in Fig. 1 and plots *D* as a function of the convection film transfer coefficient from the panel surface. Points are noted on the curve corresponding to the values of the coefficient which would exist if the panel were installed in floor, wall and ceiling.

The construction of the curve was based on constant panel area with consequent reduction in surface temperature as the convection increases. For the three points marked on the curve the respective surface temperatures are 120 F (ceiling), 112 F (wall) and 104 F (floor). Referring to Fig. 3, the drop in *D* corresponding to a reduction in ceiling panel temperature from 120 F to 104 F would be less than 1 F. From Fig. 5, however, the drop is 9 F so that the influence of panel location is obviously very much greater than the effect of the temperature change.

If performance of a panel system is measured in terms of the fraction of energy transferred by radiation, that is, in terms of the magnitude of *D*, then it follows directly that ceiling panels come closest and floor panels farthest from the ideal radiant heating system. A well designed floor panel installation may equal in performance a convection heating system and may possess other advantages which justify its use, but it does not in any marked degree possess the characteristics which are said to be desirable in radiant panel heating.

Conclusions

The trend curves presented in this paper were established for a particular structure by application of the rational equations developed in a previous paper. While numerically significant only for the conditions used in their construction, the curves give certain general qualitative results which can be taken as applicable to any problem involving design or analysis of a panel heating system.

Taking the depression *D*, of air temperature below mean radiant temperature as a criterion of effectiveness in radiant heating, five conclusions follow from the curves:

1. Greatest advantage is realized from panel heating in localities where the winter outside temperature is low; effectiveness increases with heating load.

2. Ventilation rate is the most important single factor influencing panel performance. At zero ventilation rate there can be no difference between mean radiant temperature and air temperature; at a rate of ½ air change per hour *D* becomes significant and at higher ventilation rates it assumes large values. The rational expression for ventilation in panel heating is cubic feet of air per unit time per unit inside surface area.

3. The temperature of a panel has but small influence on its performance; in the range from 90 to 150 F increased surface temperatures give a slightly greater *D*.

4. Effectiveness increases as insulation decreases. This does not mean that insulation is undesirable, but simply that the advantage of panels is greater in poorly constructed buildings.

5. Effectiveness is greatly reduced if the panel is placed in the wall or floor rather than the ceiling. Floor panels have such a small *D* that their performance can best be estimated by considering them as a form of convective rather than radiant heating system.

Summarizing the foregoing conclusions in terms of the variables which appear in the rational equations: The ceiling is the most effective location for radiant heating panels and the performance of such panels becomes more effective as out-

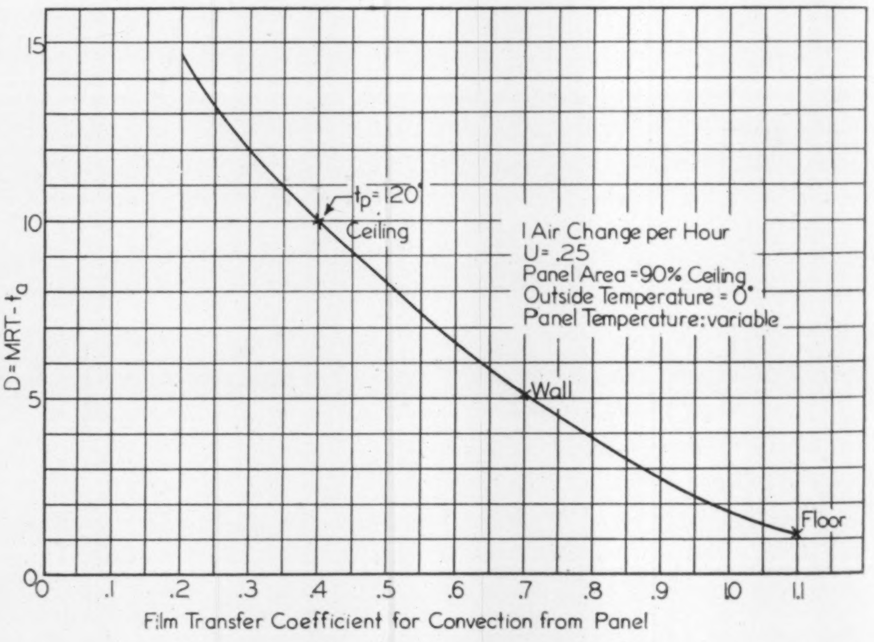

Fig. 5—Panel effectiveness *vs.* panel location

side temperatures decrease and as inside temperature, panel surface temperature, ventilation rate, and structure conductance increase.

Appendix

The equations used in establishing the trend curves presented in this paper are simplified forms of the basic rational equations derived in an earlier paper[4]. Except where otherwise noted, the nomenclature for the following equations corresponds to that used in the original publication.

Equation I—Heat balance on unheated wall surface: Equation (48) of the original paper established a heat balance on the unheated section of wall surface. This equation was,

$$t_w = \frac{h_R(F_{fw}t_f + F_{1w}t_1 + F_{pw}t_p + F_{cw}t_c + F_{gw}t_g) + h_w t_a + C_w t_o}{h_R(F_{fw} + F_{1w} + F_{pw} + F_{cw} + F_{gw}) + h_w + C_w} \ldots\ldots (48)$$

For the room considered in construction of the trend curves all inside surface is either heated or is unheated surface having constant conductance. The panel area is considered small with respect to the size of the enclosure so that $F_{wp} = 1.0$ and, by the reciprocity theorem, $F_{pw} = A_p/A_w$. Letting $h_R = 1.0$ and $h_w = 0.80$ Equation (48) then becomes,

$$t_w = \frac{A_p t_p + 0.8A_w t_a + C_w A_w t_o}{A_p + 0.8A_w + C_w A_w} \ldots (a)$$

Letting $A_p/(A_p + A_w) = u$ and $A_w/(A_p + A_w) = v$ where u and v are then the fractions of total inside surface which are heated and unheated respectively Equation (a) becomes,

$$t_w = \frac{ut_p + 0.8vt_a + C_w v t_o}{u + 0.8v + C_w v} \text{ (Equation I)}$$

Equation II—Heat balance on air passing through room: Equation (55) of the original paper establishes a heat balance on the ventilation air passing through the room,

$$t_a = \frac{h_p t_p A_p + h_c t_c A_c + h_g t_g A_g + h_w t_w A_w + h_f t_f A_f + h_1 t_1 A_1 + 0.24W t_o}{h_p A_p + h_c A_c + h_g A_g + h_w A_w + h_f A_f + h_1 A_1 + 0.24W} \ldots (55)$$

Simplifying this equation by the same substitutions used in *Equation I* with the added substitution of,

$V_c = \frac{13.5W}{A_p + A_w}$ where V_c is then the cubic feet of ventilation air passing each square foot of inside surface per hour,

$$t_a = \frac{0.4ut_p + 0.8vt_w + 0.0178V_c t_o}{0.4ut_p + 0.8t_w + 0.0178V_c} \text{ (Equation II)}$$

Equation III—Comfort Equation: Fig. 1 of the original paper presented a comfort line defined by the equation,

$$t_a = \varnothing\ (\text{MRT})$$

Based on the assumptions discussed in an earlier section of this paper the comfort line is considered straight, of unit slope and passing through the point 72F, 72F,

$$t_a = 144 - \text{MRT}$$

but, $\text{MRT} = ut_p + vt_w$

$$\therefore t_a = 144 - ut_p - vt_w \quad \text{(Equation III)}$$

Analysis

Equations I, II, and III independently relate the seven variables: t_p, t_w, t_a, t_o, C_w, V_c and u. From three simultaneous equations it is possible to eliminate two variables, as t_a and t_w, to give an equation of the form,

$$u = \varnothing\ (t_p, t_o, C_w, V_c). \quad \text{(Equation IV)}$$

Fig. 1: Using *Equation IV* with $t_p = 120$ F; $t_o = 0$ F, $C_w = 0.29$ and $V_c = 2.5$ (corresponding to one air change per hour in a room 20 ft x 20 ft x 10 ft) a solution for the panel area becomes possible, yielding $u = 0.225$ (90 per cent of ceiling area in 20 ft x 20 ft x 10 ft room). Then substituting known values of u, V_c, and C_w in *Equation IV* to obtain,

$$t_p = \varnothing\ (t_o) \quad \text{(Equation V)}$$

Equations I, II, and III can be solved to obtain an equation of the form, $D = \varnothing\ (u, C_w, V_c, t_p, t_o)$ where all variables except t_p and t_o are constants and these two terms are related by *Equation V*. Substituting,

$$D = \varnothing\ (t_p)$$

which is the relationship shown in Fig. 1 of the paper.

By a similar method of analysis other factors can be eliminated and the equations established to permit plotting Figs. 2, 3, 4 and 5.

All points on all curves satisfy the three basic panel analysis equations and therefore represent state points corresponding to comfort (as defined in *Equation III*) as well as equilibrium in terms of the heat transfer characteristics of the system.

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[4] Loc. Cit., Note 1.

COLUMBIA TO INVESTIGATE HEAT FLOW THROUGH WALLS

A cooperative agreement to investigate the heat flow through various building walls has been announced by F. C. McIntosh, Pittsburgh, Pa., Chairman of the ASHVE Committee on Research, between the Society and School of Engineering, Columbia University, New York. The work will be carried on in the heat transfer laboratory at the University with a specially designed apparatus for measuring transient heat flow. The results are expected to enable the engineer to have additional data available for analyzing more accurately the cooling requirements of an air conditioned building.

RALPH GONZALEZ COMMISSIONED IN ORDNANCE DEPARTMENT

R. A. Gonzalez, one of Chrysler Airtemp's engineers, has left Dayton to accept a Captain's commission in the Ordnance Department of the U. S. Army. He joined the Airtemp organization in 1935, coming to Dayton from York Ice Machinery Corp., where he had long been engaged in air conditioning sales work. An instructor in air conditioning for the York organization, he continued this work during the first years at Airtemp, training many of the men now with the company in the field.

A native of San Antonio, Tex., he was graduated from A. and M. College of Texas, known for its engineering graduates and fine military training. From that time until now he held a commission in the U. S. Army Reserve.

Overloading of Viscous Air Filters During Accelerated Tests

By Frank B. Rowley* and Richard C. Jordan**, Minneapolis, Minn.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the Engineering Experiment Station, University of Minnesota

SUMMARY—Dry filters have no tendency to overload during accelerated tests, but with oil filters overloading may occur if the rate of dust feed is more rapid than the rate with which the adhesive can absorb the dust. Tests conducted on viscous filters with comparatively light adhesives showed no appreciable change in efficiency with temperatures between zero and 130 F. By tests conducted on various carbon and lamp blacks, it was found that a dust to possess non-overloading characteristics when used as a component of a filter test dust should have a high rate of oil absorption but a low weight of oil absorption. One dust mixture tested was found capable of duplication and possessed non-overloading characteristics

AIR FILTERS of the unit replaceable type may be classified according to their principle of operation as either viscous or dry. The viscous type of filter usually consists of a fibrous or cellular material coated with an adhesive. The filter pack is usually graded as to density, fiber size, adhesive coating, or some combination of these three for the purpose of progressively cleaning the air as it passes through the filter and for distributing the load throughout the filter. Dust is eliminated from the air by subdividing the air stream in its passage through the filter to increase the possibility that all of the dust particles may come into contact with the adhesive material. The air-dust mixture is made to change directions frequently in its passage through the filter, and the dust is thus centrifuged against the oiled surface.

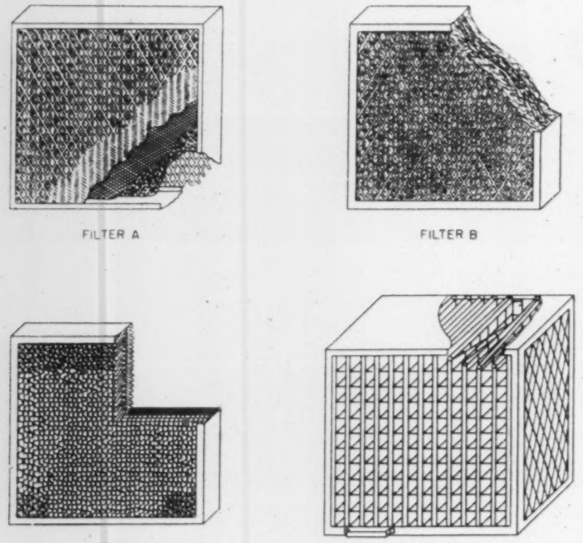

Fig. 1—Sectional views of air filters tested

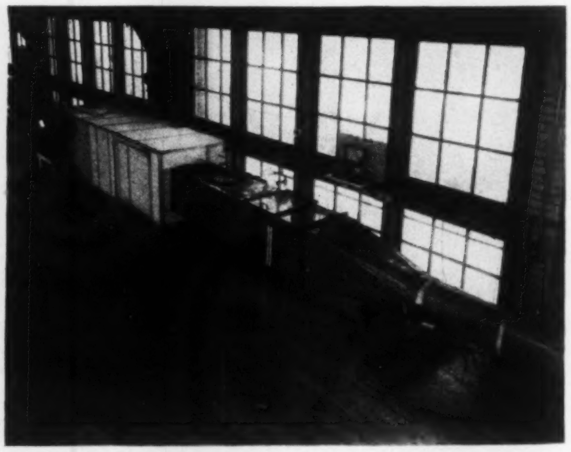

Fig. 2—View of air filter test apparatus showing air temperature control chamber

The dry type of filter works upon a somewhat different principle from that of the viscous type. It generally consists of a finely woven or packed material having comparatively small interstices. In some dry filters the air passages are sufficiently fine that the dust is removed by the screening action. However, it appears that in most cases the major part of the dust is removed as in the viscous filters by impingement and centrifuging.

In the laboratory rating of air filters by the weight or crucible method several difficulties are introduced through the necessity that the tests be accelerated. Since it would be impractical to feed an artificial test dust to a filter in the laboratory at the same rate that it would be subjected to dust in an actual installation, the test filters are subjected to rates of feed 40 to 160 times normal in order that the test may be completed in a reasonable period of time. This requirement of a practical laboratory test apparently has little effect upon the performance characteristics of a dry type of filter as the efficiency and dust holding capacity vary only slightly from low to high rates of dust feed. With the viscous type of filter, however, it is possible to feed the dust to the filter faster than the adhesive can absorb the impinged dust particles. The limiting rate of feed during an accelerated test is apparently dependent not only upon the nature of the adhesive but also upon the design of the filter and the physical properties of the dust be-

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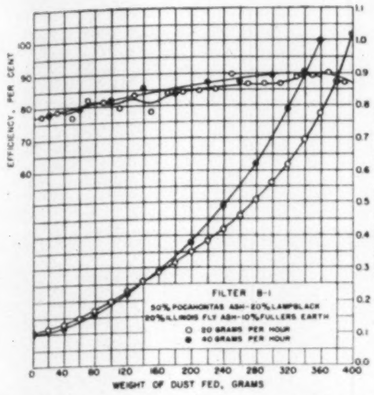

Fig. 3—Performance curves for *Filter B-1* at two rates of dust feed

ing used for test purposes. Thus a rate of dust feed which will cause overloading with one type of filter and one type of dust may be satisfactory for the testing of other filters and with other test dusts. The results of overloading are to reduce the average efficiency of the filter and usually, but not always, to increase the life and dust-holding capacity. For these reasons there has been a need for extended studies covering the causes of filter overloading and means for its prevention. The present investigations cover the following points:

1. The test conditions under which filter overloading may occur.
2. The physical properties of test dusts which cause filter overloading.
3. Dust components which are readily reproducible and which will not result in fictitious performance characteristics through overloading.

Test Dust Mixtures

One of the major problems in the formulation of a standard air filter code for the rating of filters by the weight method has been the determination of a satisfactory artificial test dust. Such a dust must not only simulate air-borne dust in all of the properties which may affect filter performance, but it must also be reproducible as to its physical properties, must not cause filter overloading at practical rates of dust feed, and must be practical to obtain in sufficient quantities for test purposes.

Various mixtures of synthetic components have been tried at different times and proposed as standard dust mixtures. The original ASHVE Air Filter Test Code specified a mixture consisting of 50 per cent by weight of powdered lampblack, containing a minimum of 97.5 per cent free carbon, and having a minimum bulking value of 3.5 lb per cubic foot, mixed with 50 per cent by weight of Pocahontas bituminous coal ash screened to pass a 200 mesh sieve. However, it was shown later[1] that many different carbons having widely varying effects upon air filter performance characteristics would satisfy this requirement. Furthermore, filters of the same type, when tested with different lots of lampblack of the same brand and manufacture, showed efficiency values ranging from 28.1 to 82.4 per cent. It was, therefore, obviously impractical to use a material with such a wide range of test results as a basis for a standard test dust. Such a dust was satisfactory only for comparative performance tests of different filters in which a sufficiently large supply of the mixture was prepared for use throughout the entire series of investigations.

A test dust consisting of 50 per cent Pocahontas ash, 20 per cent Illinois fly ash, 20 per cent lampblack, and 10 per cent Fuller's Earth was selected as a basis for a series of railroad air filter tests made at the University of Minnesota in 1937. This dust was the result of a conference held by various manufacturers and users of air filters. This air filter test dust too was satisfactory only for comparative tests as no rigid specifications were set upon the physical properties of the components, and therefore, there was no assurance that the dust would be reproducible at a later date or in some other section of the country. A great many other dust mixtures have been proposed at various times, but without exception the stipulations placed upon the physical properties of the components have been inadequate to insure their satisfactory use as a standard test dust.

In 1939 a series of tests was conducted at the Engineering Experiment Station of the University of Minnesota in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in order to investigate various materials which might be synthesized and used satisfactorily as components of an artificial test dust. During these tests 15 different types of materials were investigated[2] and as a result it was tentatively recommended that one dust mixture

Table 1—Comparison of Filter Performance Characteristics for Overload Tests

Filter	Per Cent	Dust Mixture	Rate of Feed, Grams/Hr	Filter Performance Based on 0.4 In. H_2O Final Resistance			Filter Performance Based on 1.0 In. H_2O Final Resistance		
				Efficiency, Per Cent	Life, Hours	Dust Holding Cap., Grams	Efficiency, Per Cent	Life, Hours	Dust Holding Cap., Grams
B-1	50 20 20 10	Pocahontas Ash Illinois Fly Ash Lampblack Fuller's Earth	20	81.1	11.5	186.6	84.0	19.9	334.4
B-1	50 20 20 10	Pocahontas Ash Illinois Fly Ash Lampblack Fuller's Earth	40	82.6	5.2	171.9	85.3	9.0	307.2
B-1	80 20	Pocahontas Ash Carbon Dust[a]	40	86.9	7.0	243.0	58.5[b]	......	....
B-1	80 20	Pocahontas Ash Lampblack	40	83.8	4.2	140.7	86.3	8.8	303.8
C	50 20 20 10	Pocahontas Ash Illinois Fly Ash Lampblack Fuller's Earth	40	72.6	10.2	296.3	70.6	18.4	519.4
C	80 20	Pocahontas Ash Carbon Dust[a]	40	47.4	21.5	408.0	41.5	44.5	738.9
C	80 20	Pocahontas Ash Carbon Dust[a]	20	70.2	16.0	224.6	66.9	32.6	436.2
D	50 20 20 10	Pocahontas Ash Illinois Fly Ash Lampblack Fuller's Earth	40	94.7	7.2	272.8	95.5	9.1	347.6
D	80 20	Pocahontas Ash Carbon Dust[a]	40	96.0	8.6	330.4	96.9	11.8	457.2

[a] Double Bolted Carbon Dust (Binney and Smith Co.)
[b] Average for 18-hour test period

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[1] ASHVE RESEARCH REPORT No. 1122—Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

[2] ASHVE RESEARCH REPORT No. 1143—A Standard Air Filter Test Dust, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

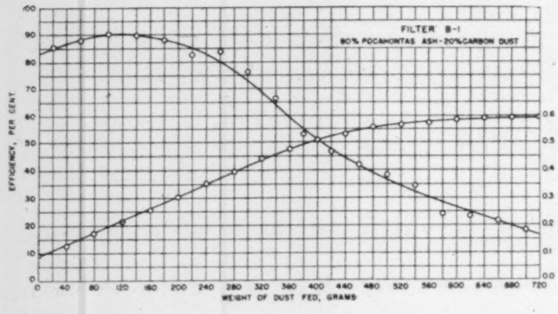

Fig. 4—Performance curves resulting from overloading of *Filter B-1*. Dust feed, 40 grams per hour

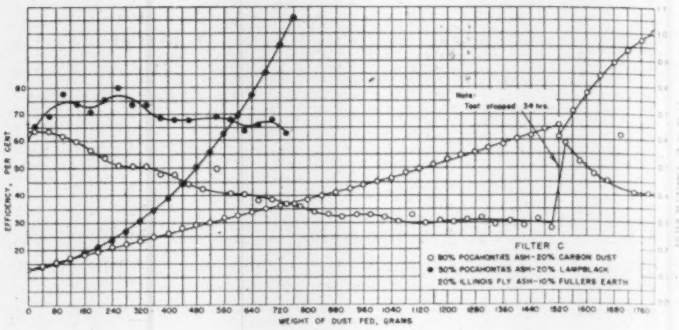

Fig. 5—Performance curves for *Filter C* on an overloading and a non-overloading dust mixture. Dust feed, 40 grams per hour

which would be reproducible and, in the main, satisfy the general requirements of a standard dust would consist of a mixture of 80 per cent by weight of Pocahontas ash screened through a 200 mesh sieve and having a jolted density between 0.65 and 0.85 grams per cubic centimeter and 20 per cent by weight of Double Bolted Carbon Dust (Binney and Smith Co.) screened through a 100 mesh sieve and having a jolted density between 0.145 and 0.180 grams per cubic centimeter. This dust was used for some time at the University of Minnesota and was found in general to satisfy the majority of the requirements of a standard dust including that of reproducibility. However, it was noted that under certain conditions of test and with certain specific filters the test results indicated that the filter was being overloaded, and therefore the test results were, to a certain extent, falsified.

Test Procedure

In order to carry out these investigations on the overloading of viscous air filters, four typical filters as shown in Fig. 1 were chosen for experimental laboratory tests. These may be described briefly as follows:

Filter A—A permanent type of cleanable oil filter, 2 in. thick, with layers of expanded metal and wire screen graded from coarse mesh at the entrance to fine mesh at leaving side.

Filter B—A viscous coated throw-away filter 2 in. thick. The fibrous media were graded in fiber size, density, and oiling from entering to leaving side. A modification of this filter, which has been designated as *Filter B-1* in these tests, was also used in some of the investigations and incorporated curly instead of straight fiber media.

Filter C—A cellular type filter 2 in. thick built in two sections, each with the axis of cells set at 45 deg to the center-line of duct and at 90 deg to each other. The cells on the entering side were of larger dimensions than those on the leaving side of the filter.

Filter D—A filter of cotton media of coarse material on the entering side and glazed on the leaving side. The filter media were accordion plaited in frame to give an area of approximately 12 times the cross-sectional area of the air stream.

The weight, or crucible, method of testing and rating air filters which has been described in detail in a previous publication[3] was used in these investigations. Throughout the investigations the filter face air velocity was maintained at 300 fpm, and the dust was fed to the filter at rates of 20 or 40 grams per hour, dependent upon the specific tests. In the majority of the investigations determinations were made of the efficiency, life, and dust-holding capacity, with the latter two characteristics based upon a limiting resistance of either 0.4 in. or 1.0 in. of water, or a resistance rise of 0.15 in. of water. In many cases both limiting resistances were used to define filter life and dust holding capacity on the same tests in order to indicate the degree of filter overloading at different dust loads.

As evidence of filter overloading during an accelerated test three different criteria were used:

1. If, during a filter test, there was a marked drop in efficiency as the test proceeded, the filter was considered to have been overloaded.
2. If a filter was tested for a period of hours and then allowed to stand without test during a second period and if, upon resuming the filter tests, there was a marked increase in efficiency and a decrease in resistance, the filter was considered to have been overloaded.
3. If two identical filters were tested at different rates of dust feed and if, after correcting for the rates of feed, there was a marked difference in the performance characteristics, then the filter tested at the higher rate of feed was considered to have been overloaded.

All three of these criteria were evidence that the dust was being fed to the filter faster than it could be absorbed by the viscous coating. However, it should be realized that the second criterion noted is evident to a small degree on practically all accelerated filter tests conducted at practical rates of dust feed and that overloading accompanying any such minor indications usually has only a slight effect upon the performance characteristics.

One series of performance tests was run to determine the effect of air temperature upon filter performance and overload. The apparatus used for these tests was identical with that used during the ordinary dust tests, with the exception that an air-mixing chamber, as shown in Fig. 2, was constructed on the inlet of the test duct. This mixing chamber was so designed that air could be, drawn directly from the room, from the outside, or through a heater coil. Thus it was possible during cold winter days to run air filter tests at temperatures ranging from 0 F to 130 F. During tests the air temperatures were measured by means of a direct-reading potentiometer and four copper-constantan thermocouples located in the air stream in front of the filter.

Overload Tests

In order to show those conditions under which overloading can be obtained, a series of performance tests was made on *Filters B-1*, *C*, and *D* with various dust mixtures. *Filter B-1* was tested at rates of feed of 20 and 40 grams per hour, using the 50-20-20-10 dust mixture containing 50 per cent Pocahontas ash, 20 per cent Illinois fly ash, 20 per cent lampblack, and 10 per cent Fuller's Earth. In addition, two tests were made on *Filter B-1* at 40 grams per hour using an 80-20 dust mixture containing 80 per cent Pocahontas ash and 20 per cent Double Bolted Carbon Dust, and an 80-20 mixture containing 80 per cent Pocahontas ash and 20 per cent lampblack. *Filter C* was tested at 40 grams per hour and 20 grams per hour using the 80 per cent Pocahontas ash and 20 per cent Double Bolted Carbon Dust mixture and also at 40 grams per hour using the 50-20-20-10 dust mixture. *Filter D* was tested at 40 grams per hour using the 50-20-20-10 dust mixture, and also with the 80 per cent Pocahontas ash, 20 per cent Double Bolted Carbon Dust mixture. All of these tests were made at a filter air face velocity of 300 fpm and to a final limiting resistance of 1.0 in. of water. The results of these tests are recorded in Table 1, and in the performance curves shown graphically in Figs. 3, 4, 5, 6, and 7.

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[3] Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed, and Air Velocity through Filter, by F. B. Rowley and R. C. Jordan. (ASHVE Transactions, Vol. 44, 1938.)

The comparison between the performance characteristics for *Filter B-1*, tested on the 50-20-20-10 dust mixture at rates of feed of 20 and 40 grams per hour, is shown in Fig. 3 and Table 1. Regardless of the high limiting resistance, there is little, if any, indication of overloading at the higher rate of dust feed when using this test dust. The average efficiencies are within 1.5 percentage units when based on either a final limiting resistance of 0.4 in. or 1.0 in. of water. Both the filter life and dust-holding capacity check as closely as can be expected considering the normal variations to be found between filters of the same design and manufacture. These tests bear out all previously reported tests[4] which have shown that there is little variation in performance characteristics of the same filter when tested with this dust at rates of feed varying from 20 to 60 grams per hour.

Fig. 4 and Table 1 show the results of the tests made on *Filter B-1* with the 80 per cent Pocahontas ash and 20 per cent carbon dust mixture at a 40 gram per hour rate of feed. These results indicate that the filter became overloaded shortly before it reached a resistance of 0.4 in. of water and that thereafter it was unable to absorb the dust as rapidly as it was fed to the filter. After passing this resistance, the efficiency curve dropped rapidly and the resistance curve levelled off at approximately 0.6 in. of water. It would have been impractical to continue this test any further, and obviously any reported performance characteristics would have been fictitious as the efficiency had dropped from a high of 90 per cent to a low of approximately 18 per cent. The average efficiency, when based upon a limiting resistance of 0.4 in. of water, was 86.9 per cent; when the test was continued to a resistance of 0.6 in. of water, however, the average efficiency dropped to approximately 58.5 per cent. No appreciable overloading took place until the filter pressure drop had exceeded 0.4 in. of water, and if the test had been concluded at that time, the results would probably have been considered satisfactory.

The results of the performance tests made on *Filter B-1*, with the dust mixture consisting of 80 per cent Pocahontas ash and 20 per cent lampblack, at a dust rate of feed of 40 grams per hour, are presented in Table 1. All previous tests had indicated that the dust component causing the overloading was the Double Bolted Carbon Dust, and for this reason lampblack was substituted for carbon dust in the 80-20 dust mixture. The results of this test indicated no overloading, and performance characteristics based on either a 0.4 in. of water or 1.0 in. of water final resistance were very close to those determined for the 50-20-20-10 dust mixture on this same filter. Apparently the Pocahontas ash, the Illinois fly ash, and the Fuller's Earth all had about the same effect on the performance characteristics, and therefore the substitution of Pocahontas ash for the Illinois fly ash and the Fuller's Earth had little effect upon the resulting performance characteristics.

The effects of overloading are clearly shown in the results obtained on *Filter C* and presented in Figs. 5 and 6 and in Table 1. The performance characteristics using the 50-20-20-10 dust mixture appeared reasonably normal and showed only a slight overloading as indicated by a small drop in the efficiency curve. However, the same filter, tested at the same rate of dust feed but with the 80 per cent Pocahontas, 20 per cent carbon dust mixture, definitely indicated that overloading was taking place almost as soon as the test had started. The 80-20 dust mixture was tested at two rates of feed, 20 and 40 grams per hour. At the higher rate of feed the test was conducted for a period of 38 hours after which there was a lapse of 34 hours in which no dust was fed to the filter. The test was then resumed and continued until the filter resistance had reached 1.0 in. of water. As indicated in Fig. 5, there was a gradual reduction in efficiency from 64 per cent to 28 per cent during the feeding of 1,520 grams of dust. After the filter had stood for 34 hours without test, the efficiency immediately rose from 28 to 59 per cent upon resuming the test, and then again dropped to 40 per cent. During the 34 hour period during which the test was stopped, the filter resistance dropped from 0.66 in. to 0.61 in. of water. Furthermore, upon resuming the test there was a marked increase in the slope of the resistance curve, giving additional proof of overloading. With both the 50-20-20-10 and the 80-20 dust mixtures fed to *Filter C* at the rate of 40 grams per hour, the initial efficiencies were approximately the same. However, after the resistances had reached 0.4 in. of water, the average efficiency in the case of the 50-20-20-10 mixture was 72.6 per cent, and in the case of the 80-20 mixture was 47.4 per cent. Based upon a final limiting resistance of 1.0 in. of water, the 50-20-20-10 mixture had an average efficiency of 70.6 per cent, and the 80-20 mixture an average efficiency of 41.5 per cent.

Fig. 6 and Table 1 show the results of the test made on *Filter C* with the 80 per cent Pocahontas ash, 20 per cent carbon dust mixture at a 20 gram per hour rate of dust feed to determine whether lowering the rate of feed would eliminate or reduce the overloading. In addition to reducing the rate of feed the test was conducted in 8-hour periods interspersed by periods of rest. Although the result was to increase greatly the average efficiency, based on both the 0.4 in. and the 1.0 in. of water limiting

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[4] Loc Cit., Note 3.

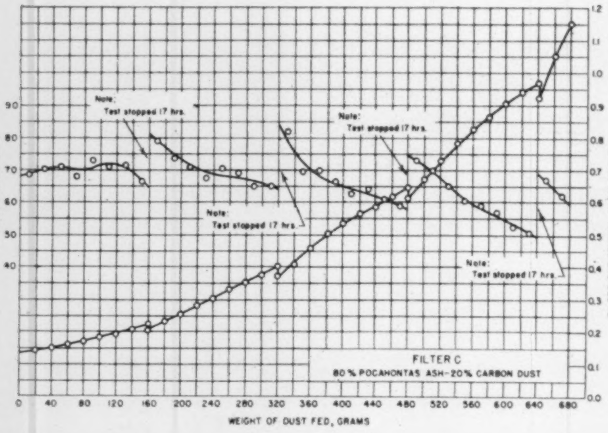

Fig. 6—Performance curves resulting from overloading of *Filter C* during an intermittent test. Dust feed, 20 grams per hour

Table 3—Air Filter Performance Characteristics at Different Air Temperatures

(*Dust mixture* 50-20-20-10. *Face Air Velocity* 300 *fpm. Dust feed* 40 *grams per hour.*)

Air Temp. Deg F	Performance Characteristic	*Filter A* (Oil No. 1)	*Filter A* (Oil No. 2)	*Filter B*	*Filter C*	
30	Efficiency, per cent	74	76	84	62	64a
	Life, hours	5.7	6.2	3.6	13.9	11.0a
	Dust Holding Capacity, grams	170	187	122	345	282a
70	Efficiency, per cent	77	80	85	72	73
	Life, hours	7.0	8.0	3.2	6.9	6.4
	Dust Holding Capacity, grams	215	256	109	198	186
130	Efficiency, per cent	78	81	85	76	74
	Life, hours	9.2	8.0	5.0	8.8	8.6
	Dust Holding Capacity, grams	286	261	168	268	254

a 20 grams per hour rate of dust feed. Life based upon 40 gram per hour rate of feed.

resistances, and to reduce both the life and dust-holding capacity, as compared with the test conducted at 40 grams per hour, there was still much evidence of overloading. During periods of rest there were marked rises in efficiency and drops in resistance, which increased in magnitude as the tests progressed. Furthermore, towards the end of the test, the curvature of the resistance graph changed from positive to negative.

The results of the performance tests on *Filter D* using the 50-20-20-10 and the 80-20 dust mixtures are shown in Fig. 7 and Table 1. Neither of these tests gave any indication of filter overloading, a result which would be expected with dry filtering media and no oil adhesive involved. The efficiency curves were practically identical, although the filter life was somewhat greater with the 80-20 dust mixture. However, variations of this magnitude are often found on supposedly identical filters tested under the same conditions.

The results of these tests indicate that filter overloading, resulting in fictitious performance characteristics, may result when viscous filters are tested under certain conditions. There is apparently no danger of overloading with dry filters, but with those designed upon the adhesive impingement principle overloading may occur if the rate of dust feed is more rapid than the rate at which the adhesive can absorb the dust. Such a condition is apparently brought about if certain types of test dusts are used, if the filter adhesives are extremely viscous and incapable of absorbing dust rapidly, or in some cases if the final limiting resistance is arbitrarily placed much beyond that to which the filter may normally be subjected under good operating conditions. It is probable that under all accelerated filter tests there is a slight overloading, but any abnormal overloading which would appreciably affect the resulting performance characteristics will be indicated either by a decrease in the efficiency as the test progresses, a sudden change in the efficiency and resistance after an interruption of appreciable duration during a test, or in a change in the performance characteristics of similar filters when tested with the same test dust but under different rates of dust feed.

Table 2 — Properties of Air Filter Adhesives

Oil No.	Filter	Pour Point, Deg F	Saybolt Viscosity, Sec.		
			70 F	100 F	130 F
1	A	10	4011	1222	486
2	A	— 5	946	320	138
3	B	20	1170	426	230
4	C	60	....	2040 (Furol tube)	805 (Furol tube at 122 F)

Filter Tests at Different Air Temperatures

A series of tests was made with *Filters A, B,* and *C* to determine the effects of air temperature upon filter performance and overloading. The dust chosen for these tests was the 50-20-20-10 mixture which had previously shown[5] no appreciable overloading on any filters tested at rates of feed varying from 2 to 100 grams per hour. In all tests the face air velocity was maintained at 300 fpm, and, with one or two exceptions as noted, the rate of dust feed was 40 grams per hour. *Filter A,* a permanent, cleanable type of filter was tested with both a light and a heavy oil, some of the properties of which are presented in Table 2 along with those for the adhesives used on the other filters tested.

One series of tests was made at air temperatures of 30, 70 and 130 F, in which the average efficiency, the life, and the dust-holding capacity were determined for a resistance rise of 0.15 in. of water. A second series of tests was conducted at air temperatures of 0, 30, 70 and 130 F with determinations made only of the average efficiency for a three-hour test period. It was impossible to obtain complete filter

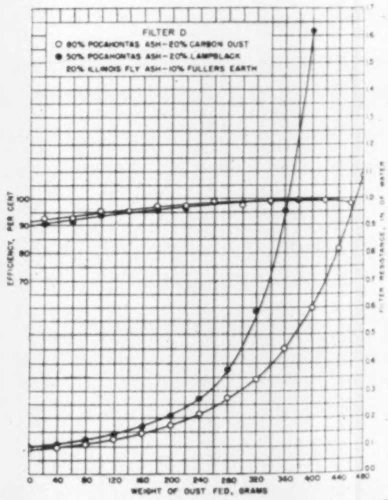

Fig. 7—Performance curves for *Filter D* on two dust mixtures. Dust feed, 40 grams per hour

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[5] Loc. Cit., Note 1.

characteristics at the 0 F temperature, as this low temperature could not be maintained over a sufficient number of hours to complete the performance tests.

Table 3 shows the complete performance characteristics determined over a 0.15 in. of water increase in filter resistance. The actual performance curves are not included because of lack of space. With the exception of *Filter C,* these tests indicated no significant differences in the efficiencies at the different air temperatures. *Filter C,* however, when tested at a rate of 40 grams per hour, dropped from an efficiency of 72 per cent at 70 F to an efficiency of 62 per cent at 30 F. In addition there was a corresponding increase in the dust-holding capacity and life at the lower temperature, and the efficiency curve showed a definite drop in efficiency as the test progressed. The 50-20-20-10 dust had on no previous occasions shown any indications of overloading even at considerably higher rates of feed, but it appeared that at this low temperature the adhesive had become so viscous that it could no longer absorb the dust as rapidly as it was fed. For this reason *Filter C* was again tested with this dust at 30 F, but at the reduced rate of feed of 20 grams per hour, in order to see if overloading could be prevented or reduced. The resulting performance characteristics showed a slightly higher average efficiency and somewhat lower life (corrected for difference in rate of dust feed) and dust-holding capacity than those at the same temperature but at the 40 grams per hour rate of dust feed. However, there was still some overloading at this 20 gram per hour rate of feed as evidenced by the lower efficiency and greater life and dust-holding capacity in comparison with the 70 F test, and by a slight drop in the efficiency curve as the test progressed.

The tests conducted on *Filter A* with oils Nos. 1 and 2 and on *Filter B* showed no indications of overloading at low temperature. Although there were slight drops in the efficiency when lowering from 70 F to 30 F, there were also decreases in life and dust-holding capacity, and such corresponding decreases are not typical of an overloaded condition. In addition, analyses of the efficiency curves showed no indications of any overloading. It would therefore appear that the differences in the performance characteristics at the different temperatures for these three filters were indicative of the true differences to be found under non-overloaded conditions.

All filters showed equal or slightly higher efficiency and higher life and dust-holding capacity at the 130 F temperature than at the 70 F temperature. This would indicate that with these filter designs somewhat better performance can be expected at the higher temperatures. *Filter B* showed the lowest life and dust-holding capacity at a temperature of 70 F, but the differences in magnitude of the performance characteristics between 30 F and 70 F were not sufficiently great to warrant any conclusion.

Table 4 presents the results of the efficiency determinations based on the three-hour tests made at the different air temperatures. Both *Filter A* with oil No. 2 and *Filter B* showed substantially the same efficiencies throughout the entire range of temperatures from 0 F to 130 F; however, *Filters C* and *A* with the more viscous adhesives showed marked reductions in efficiencies at 0 F over those determined at 30 F. The test on *Filter C* was made at a 20 grams per hour rate of dust feed and should therefore be compared only with the test made at 30 F at the same rate of feed. The efficiency of *Filter C* dropped from 68 per cent to 57 per cent and that for *Filter A* with oil No. 1 from 75 per cent to 65 per cent when the temperature was reduced from 30 F to 0 F. As it appeared probable from the first series of air temperature tests that the reductions in efficiency for *Filter C* at 30 F over the tests at 70 F were the result of overloading, these further reductions in efficiency at 0 F were probably also caused by overloading. However, there is no direct evidence to substantiate this, as it was impossible to conduct the tests over a sufficiently long period of time to determine the life and dust-holding capacity at these low temperatures. The tests on *Filter A* with oil No. 2 and on *Filter B* indicate that filtering efficiency remains constant over a comparatively wide range of temperatures if there is no overloading caused either by an accelerated rate of feed or by a filter oil too viscous to absorb the dust readily.

Table 4—Air Filter Efficiency at Different Air Temperatures

(Dust mixture* 50-20-20-10. *Face Air Velocity* 300 *fpm. Dust feed* 40 *grams per hour.)

Air Temp. Deg F	*Filter A* (Oil No. 1)	*Filter A* (Oil No. 2)	*Filter B*	*Filter C*	
0	65	75	85		57[a]
30	75	76	84	65	68[a]
70	77	78	85	71	70
130	78	79	84	72	70

[a] 20 grams per hour rate of dust feed.

As a result of these filter tests conducted at various air temperatures, it may be concluded that viscous filters with comparatively light oils will show no appreciable change in efficiency with temperature when operating between 0 F and 130 F. From the limited tests conducted it would appear that with all viscous filters there is a tendency toward greater life and dust-holding capacity when operating at high temperatures. Those filter oils showing a marked decrease in viscosity at low temperatures will result in a certain degree of overloading at low temperatures even with test dusts which give no indication of overloading at normal temperatures. Thus it would appear highly probable that any viscous filter subjected to an accelerated test with any dust mixture would show overloading if the temperature were lowered sufficiently.

Investigations of Carbons

From the previous investigations it was apparent that one of the principal components of the test dusts causing filter overloading during the accelerated tests was carbon. For this reason it was decided that more extensive investigations should be carried out for the purposes of determining those physical properties resulting in overloading and of finding a standard product which would neither vary in its physical properties from lot to lot nor result in filter overloading during test. Letters were first written to 16 different manufacturers of carbon blacks and lampblacks, explaining the problem to each of them and asking them to submit samples which, in their opinion,

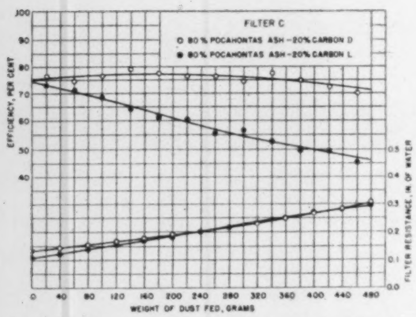

Fig. 8—Typical performance curves for *Filter C* on dust mixtures containing different carbons. Dust feed, 40 grams per hour

would be the most likely to satisfy the requirements of a carbon to be used in a test dust for air filter ratings. As a result of this correspondence, 14 different carbons were submitted for test by eight different manufacturers. Each of these carbons was given a code letter, as recorded in Table 5 along with the manufacturer's name and designation.

In order to investigate the desirable physical properties of a carbon satisfactory for air filter test purposes, two series of tests were made: one to determine the physical properties of the carbons and the other to determine the degree to which these carbons resulted in overloading when used in test dust mixtures. The physical properties investigated were the density, the rate of oil absorption, and the weight of oil absorbed. The procedures used in determining each of these properties were as follows:

Density: The jolted density in grams per cubic centimeter was determined for each of the dusts by placing it in a cylindrical container and jolting by hand to maximum density. From measurements of the weight and volume, this jolted density was then calculated.

Weight of Oil Absorbed: Several different standard tests were tried in order to determine the weight of oil absorbed in grams of oil per 100 grams of dust. The test which appeared to be the most easily duplicated and which gave the most consistent results was one developed during these investigations. Two grams of the dust to be tested were weighed and placed in a 250 ml beaker, and refiltered linseed oil with an acid value of 2-4 was then slowly added from a burette. During this process the beaker was rotated vigorously, and from time to time the addition of oil was stopped in order to make sure that sufficient time was being allowed for maximum absorption. The approach of the end point was evident by the complete disappearance of all loose dust and the formation of one large lump or ball. The weight of oil absorbed in grams oil per 100 grams of dust was then calculated from the original weight measurements of the dust and readings of the oil burette.

Rate of Oil Absorption: It was evident that, in addition to density and the weight of oil absorbed by the test dust, the rate at which the dust absorbed the oil was also an important factor in these investigations of the overloading properties of dusts. A test was therefore devised in which 20 ml of the oil to be used was placed in a 1 in. by 8 in. test tube and ½ gram of the dust under investigation added in a layer above the top surface of the oil. The tube was then placed in a rack and allowed to stand undisturbed until the end point, taken as the complete wetting of the dust by the oil, was reached. Although this test was obviously somewhat affected by the density of the dust and, therefore, the height of the column of dust above the oil, this factor is also encountered during actual filter tests. In order to simulate actual conditions as closely as possible, this rate of absorption test was conducted with the adhesive used on *Filter C* and the No. 2 oil used on *Filter A* (see Table 2).

To determine the degree to which these carbons were satisfactory for actual test purposes, a series of investigations was made with dust mixtures consisting of 80 per cent by weight of Pocahontas ash screened through a 200 mesh sieve and 20 per cent by weight of the particular carbon under investigation screened through a 100 mesh sieve. *Filter C* was chosen for these tests, as the previous investigations had shown it to be one of the most susceptible to overloading. Filter tests were made at a face air velocity of 300 fpm and at a 40 grams per hour rate of dust feed with each of these dust mixtures. Each test was continued until 480 grams of the dust had been fed to the filter, and from the data collected, the average efficiency, the maximum efficiency, the resistance rise, and the dust-holding capacity were all computed. In addition the drop from maximum to minimum efficiency was recorded and used as a criterion of the degree of overloading.

The results of the tests to determine the physical properties of the different carbons, all screened through 100 mesh, are presented in Table 6. The dust designation is shown in the first column, and the jolted density in grams per cubic centimeter and weight of oil absorbed in grams oil per 100 grams of dust are recorded in the second and third columns. The rates of oil absorption, using the oils from *Filters A* and *C*, are shown in the last two columns. The jolted densities varied widely, ranging from 0.07 to 0.48 grams per cubic centimeter. The weights of oil absorbed per 100 grams of dust varied from 63 to 796 grams. The rates of oil absorption also varied widely between the different dusts, and, in addition, varied considerably be-

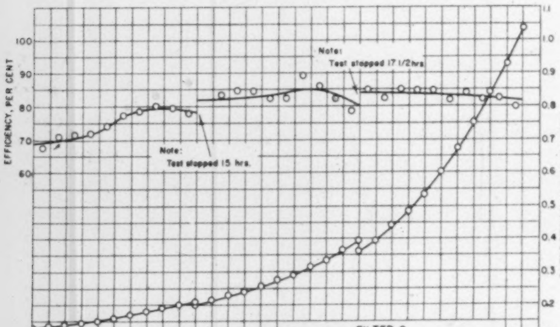

Fig. 9—Performance curves for *Filter C* on 80 per cent Pocahontas ash, 20 per cent K-1 black. Dust feed, 20 grams per hour

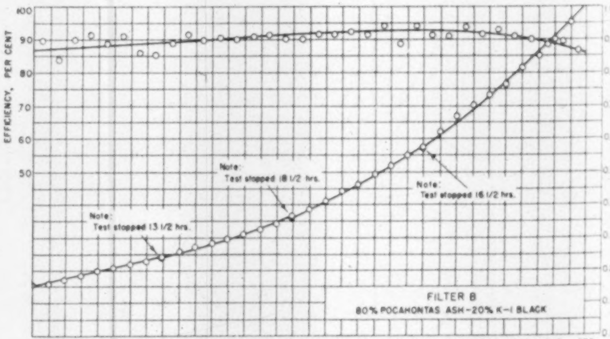

Fig. 10 — Performance curves for *Filter B* on 80 per cent Pocahontas ash, 20 per cent K-1 black. Dust feed, 20 grams per hour

Table 5—**Carbon Blacks and Lampblacks Submitted for Test**

CARBON	MANUFACTURER	MANUFACTURER'S DESIGNATION
A	Binney and Smith Co.	Raven No. 15 Carbon Black
B	Binney and Smith Co.	Molacco Carbon Black
C	Binney and Smith Co.	Double Bolted Carbon Dust
D	L. Martin Co.	K-1 Black
E	Samuel Cabot, Inc.	Processed No. 8 Lampblack
F	G. L. Cabot, Inc.	Carbolac 2 Carbon Black
G	G. L. Cabot, Inc.	Elf 30 Carbon Black
H	Monsanto Chemical Co.	No. 10 Lampblack
I	General Atlas Carbon Co.	No. 9010 Black
J	General Atlas Carbon Co.	No. 9011 Black
K	General Atlas Carbon Co.	No. 12509 Black
L	General Atlas Carbon Co.	No. 12511 Black
M	Bihn and Wolff Co.	No. 1012 Lampblack
N	Imperial Oil and Gas Products Co.	Excello Carbon Black

Table 6—**Density and Oil Absorption Tests of Different Dusts**

DUST	JOLTED DENSITY, GRAMS PER CU CM	OIL ABSORPTION, GRAMS OIL PER 100 GRAMS DUST	RATE OF OIL ABSORPTION (OIL NO. 4) GRAMS PER HOUR x 10^3	RATE OF OIL ABSORPTION (OIL NO. 2) GRAMS PER HOUR
Carbon *A*	0.22	171	10.6	1.2
Carbon *B*	0.39	72	19.4	6.3
Carbon *C*	0.16	217	3.6	0.6
Carbon *D*	0.48	63	24.6	7.1
Carbon *E*	0.23	140	7.3	2.0
Carbon *F*	0.12	424	1.5	0.5
Carbon *G*	0.17	183	6.7	0.8
Carbon *H*	0.25	116	9.4	1.4
Carbon *I*	0.28	102	10.6	1.6
Carbon *J*	0.21	150	7.3	1.0
Carbon *K*	0.17	203	5.8	0.9
Carbon *L*	0.08	796	0.5	0.9
Carbon *M*	0.07	289	1.2	0.2
Carbon *N*	0.24	164	7.3	1.4
Pocahontas Ash	0.77	66	62.1	20.0
Illinois Fly Ash	0.74	64	85.6	25.8
Fuller's Earth	0.60	80	73.0	12.9

Table 7—Comparison of Performance Characteristics on *Filter C* with 80-20 Dust Mixtures Containing Different Carbons (*Face Air Velocity* 300 *fpm. Dust rate of feed* 40 *grams per hour. Weight Dust Fed* 480 *grams*)

DUST	AVERAGE EFFICIENCY, PER CENT	RESISTANCE RISE, INCHES OF WATER	DUST COLLECTED IN FILTER, GRAMS	MAXIMUM EFFICIENCY, PER CENT	EFFICIENCY DROP, PERCENTAGE UNITS
A	61.2[a]	0.19[a]	294[a]	70	20[a]
B	72.9	0.14	350	77	7
C	54.1	0.16	260	73	20
D	75.5	0.18	363	77	7
E	68.0	0.25	326	71	10
F	53.4	0.12[a]	256[a]	65	22[a]
G	63.3	0.18	304	70	20
H	70.6	0.23	339	76	11
I	63.1	0.20	303	72	20
J	61.1	0.18	293	73	28
K	59.3	0.18	284	68	24
L	59.0	0.19	283	73	28
M	64.5	0.33	309	74	19
N	60.2	0.15	289	72	27

[a] Approximately 400 grams dust fed and results extrapolated.

Table 8 — Rank Difference Correlation Coefficients Between Efficiency Drop, Weight of Oil Absorbed, Rate of Oil Absorption and Dust Density

	WEIGHT OF OIL ABSORBED	JOLTED DENSITY	EFFICIENCY DROP	RATE OF ABSORPTION (OIL NO.2)
Rate of Absorption (Oil No. 4)	0.93	0.93	0.53	0.84
Rate of Absorption (Oil No. 2)	0.91	0.95	0.50	
Efficiency Drop	0.60	0.52		
Jolted Density	0.96			

tween the two oils, with the adhesive from *Filter C* requiring an average of approximately 180 times as long to absorb the dust as the adhesive from *Filter A*. This slow rate of absorption for the *Filter C* adhesive probably explains the marked tendency for this particular filter to overload under accelerated test.

In addition to the determinations made of the density and oil absorption properties of the different carbons, similar tests were made on Pocahontas ash, Illinois fly ash and Fuller's Earth, all screened through a 200 mesh sieve. These dusts, in comparison with the carbons, all showed high density and rate of absorption and a low weight of oil absorbed. As none of these dusts had ever shown any tendency toward causing filter overloading, it was considered probable that such properties were necessary for a non-overloading dust.

The results of the tests conducted on *Filter C* with the 80-20 mixtures of Pocahontas ash and the different carbons are shown in Table 7. Typical performance curves for two of these dust mixtures are presented in Fig. 8. These tests show a wide variation in the efficiency drop during the tests for the various dusts, the maximum being 28 per cent and the minimum seven per cent. If it had not been for the varying degrees of overloading experienced with the different test dusts, it appears likely that the average efficiencies for all tests would have been nearly the same, as the maximum efficiencies were all between 70 and 77 per cent with only two exceptions. As would be expected with varying degrees of overloading for the different tests, there were wide variations in the values for dust-holding capacity and for resistance rise.

In order to determine those physical properties of the carbons which had the greatest influence upon filter overloading, rank difference correlation coefficients were calculated for each pair of the following factors: rate of absorption (filter oil No. 4), rate of absorption (filter oil No. 2), weight of oil absorbed, jolted density, and efficiency drop. These correlation coefficients are presented in Table 8 and were based upon giving the best rankings to the highest rate of absorption, the lowest weight of absorption, the lowest efficiency drop, and the highest density. The highest or lowest values were ranked best for each factor, dependent upon which appeared to be preferable as far as preventing filter overloading. In interpreting these results, it would appear that all of the coefficients were of sufficient magnitude to indicate statistically significant results. The correlations between any two of the factors, rate of absorption (oil No. 2), rate of absorption (oil No. 4), jolted density, and weight of oil absorbed, were all very high. This indicated that it made little difference whether an oil of low or high viscosity was used in determining the relative rankings of the dust as to absorption, and that a carbon dust which would absorb comparatively small weights of oil would also absorb it rapidly. A dust which had a high density had a high rate of oil absorption, but absorbed only a comparatively small amount of oil. Although the correlations between efficiency drop and rate of absorption, weight of oil absorbed, and density were not as marked as in the other cases, they were, however, of sufficient magnitude to be significant. Furthermore, nine out of the fourteen filter tests using the 80-20 dust mixtures with different carbons resulted in efficiency drops between 19 and 28 percentage units. Because of this clustering, it appears quite probable that if the tests had been conducted over longer periods of time, they would have resulted in more marked differences in efficiency drops between the various dusts and therefore very possibly in higher correlations. Nevertheless, this analysis indicates that those carbons having the highest rates of oil absorption and the lowest weight of oil absorbed will result in the least amount of overloading. Although these correlations were made only with different blacks, it is interesting to note that Pocahontas ash, Illinois fly ash, and Fuller's Earth have all been used at various times either as components of test dusts or as test dusts themselves without resulting in any filter overloading, and that, as shown in Table 6, these dusts also are all of comparatively high density, high rate of oil absorption, and low weight of oil absorbed. The two carbons, *B* and *D*, which, when mixed with Pocahontas ash resulted in the lowest efficiency drops during the filter tests, both had the highest densities, the lowest weights of oil absorbed, and the highest rates of oil absorption of any of the blacks tested. These two dusts were respectively Molacco Black and K-1 Black. (See Table 5.)

Non-Overloading Dust Mixtures

The results of the filter tests of different carbons and of the 80 per cent Pocahontas and 20 per cent carbon mixtures indicated the physical properties of dusts possessing satisfactory non-overloading characteristics for accelerated filter tests. However, in addition to these characteristics, any such dusts satisfactory for a standard mixture must not vary appreciably in physical properties from batch to batch during manufacture. There are probably many different dusts which would meet most of the requirements of a satisfactory mixture; there are also probably many different ratios of these dust components which would be satisfactory. The overload tests already discussed indicated that one such mixture was that composed of 80 per cent Pocahontas ash and 20 per cent K-1 Black. However, in order to establish fully that this was the case, two additional tests were run with this mixture on *Filters C* and *B* at a rate of dust feed of 20 grams per hour and to a final limiting resistance of 1 in. of water. These tests were run intermittently with time lapses allowed between each eight hours of actual test. In addition eight different four-hour tests were run on *Filter B* using seven different samples of K-1 Black obtained from different production batches at different dates. The rate of dust feed was 20 grams per hour and no other components were added to the test dust.

Table 9—Comparisons of Black D[a] on *Filter B*

(*Dust Feed* 20 *grams per hour. Face Air Velocity* 300 *fpm*)

Test No.	Sample	Jolted Density, Grams per cu cm	Average[b] Efficiency, Per Cent	Resistance[b] Rise In. of H_2O
1	(a)	0.437	82.7	0.056
2	(b)	0.434	80.9	0.066
3	(c)	0.441	81.0	0.043
4	(d)	0.408	75.5	0.055
5	(e)	0.443	88.5	0.047
6	(f)	0.464	81.4	0.044
7	(g)	0.428	81.0	0.052
8	(g)	0.428	75.0	0.064

[a] All same brand and same manufacture but different production batches.

Based on 4-hour average.

Table 9 presents the results of the four-hour tests made on *Filter B* with K-1 Black and records the sample designation, the jolted density, the average efficiency, and the resistance rise. The densities for the different samples varied from 0.428 to 0.464 grams per cubic centimeter and the average efficiencies for the four-hour tests ranged from 75.0 to 88.5 per cent. Much of the variation in efficiency may be attributed to differences in the filters even though of the same brand and manufacture. For example, dust sample *G* was tested on two supposedly identical filters, and yet the average efficiencies for these tests were 75.0 and 81.0 per cent. In view of the results obtained on other carbons with similar tests at a previous date,[6] these results may be considered very satisfactory from the standpoint of duplication.

Figs. 9 and 10 are the performance curves for the tests conducted on *Filters C* and *B* with the 80 per cent Pocahontas, 20 per cent K-1 Black mixture to a limiting resistance of 1 inch of water. Each of these tests was run in 8-hour increments with time lapses as noted on the graphs. The test on *Filter B* showed practically no indication of overloading with only slight changes in resistance and no noticeable changes in efficiency between test periods. The test on *Filter C* indicated a slight overloading as evidenced by changes in resistance and efficiency during the intervening time periods. A similar test run on *Filter C* with this dust mixture at 20 grams per hour but with continuous instead of intermittent testing showed no indication of overloading, although of course slight overloading was probably taking place. In the light of previous research on dust mixtures it appears that regardless of the mixture, a filter utilizing an adhesive as viscous as that on *Filter C* will evidence a slight overloading during accelerated tests. However, a slight overloading apparently has little, if any, effect upon the efficiency and only a small effect upon the life and dust-holding capacity. These tests therefore indicate that the dust mixture consisting of 80 per cent Pocahontas ash, screened through 200 mesh, and 20 per cent K-1 Black, screened through 100 mesh, is one mixture which satisfies the requirements as to reproduction of components and as to non-overloading characteristics required of a standard dust mixture.

Summary

In this paper it has been shown that there is apparently no danger of filter overloading during accelerated tests of dry filters, but that with oiled filters overloading may occur if the rate of dust feed is more rapid than the rate with which the adhesive can absorb the dust. If such is the case, overloading may result in fictitious performance characteristics usually showing a decrease in average efficiency and an increase in life and dust holding capacity over normal performance. Such a condition can be brought about if the filter adhesive is extremely viscous and incapable of absorbing dust rapidly, if the final limiting resistance is arbitrarily placed much beyond that to which the filter may normally be subjected under good operating conditions, or if low density test dusts having a low rate of oil absorption and a high weight of oil absorption are used in the test mixture. Overloading can be detected during accelerated tests either by a drop in efficiency as the test progresses, a sudden change in efficiency and filter resistance after an interruption of appreciable duration during a test, or by a change in the performance characteristics of like filters when tested with the same dust mixture but at different rates of feed.

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[6] Loc. Cit., Note 1.

Tests conducted on viscous filters with comparatively light adhesives showed no appreciable change in efficiency with temperature when operating between 0 F and 130 F. However, those filters with oils having a marked decrease in viscosity showed a certain degree of overloading at low temperatures even with a test dust which gave no indication of overloading at normal temperatures. There appeared to be a definite tendency toward greater life and dust-holding capacity at high temperature for all viscous filters tested.

As a result of tests conducted on various carbon blacks and lampblacks submitted by different manufacturers, it was found that a dust, to possess satisfactory non-overloading characteristics when used as a component of a filter test dust, should exhibit a high rate of oil absorption, but a low weight of oil absorption. In addition, any dusts satisfactory for use in a standard test mixture must be capable of duplication. One dust mixture tested which satisfied these requirements as to reproduction of components and non-overloading characteristics consisted of 80 per cent Pocahontas ash screened through 200 mesh and 20 per cent K-1 Black screened through 100 mesh.

MELCHETT MEDAL TO A. C. FIELDNER

The Council of the British Institute of Fuel has announced that A. C. Fieldner, Chief of the Technologic Branch and Chief of the Coal Division, U. S. Bureau of Mines, will be the recipient of the Melchett Medal. This medal, awarded annually by the Institute for outstanding achievement in work involving the scientific preparation or use of fuel, is ordinarily presented personally to the recipient in London, England, in October, but under present conditions, the Council of the Institute of Fuel will endeavor to have the formal presentation made in the United States this year.

Mr. Fieldner, a chemical engineer and scientific research worker for 35 years in the Department of the Interior, thus will become the second American so honored since the medal was first awarded in 1930, the first having been given to the late Charles M. Schwab.

A native of Ohio where he attended the public schools and Ohio State University, Columbus, Mr. Fieldner launched his successful career in scientific fuel research soon after he received his college degree. He spent two years as an industrial fuel engineer and chemist with private industry, and then joined the Government service in 1907 as chemist in the Technologic Branch of the United States Geological Survey. Two years later, at the age of 27, he became a member of the staff of the U. S. Bureau of Mines Fuels Chemistry Laboratories and after a short time was placed in charge of that work.

During World War 1, Mr. Fieldner was in charge of gas mask investigations, an activity conducted by the Bureau of Mines until the middle of 1918 when it was taken over by the War Department. He served with the Chemical Warfare Service with the rank of Major from July 1, 1918 to January 11, 1919, directing the Gas Mask Research Division. During this war period, he developed methods of testing gas masks and gas absorbents and directed the research activities of more than a thousand scientists in this branch of chemical warfare.

A. C. Fieldner

In 1919 he was made supervising chemist of the Bureau and later was also appointed Superintendent of the Experiment Station at Pittsburgh, Pa. Outstanding work won for him successive promotion to Chief Chemist, Chief Engineer of the Experiment Stations Division with headquarters in Washington, D. C., and in 1936, to the position which he now holds as head of the Coal Division and Chief of the Technologic Branch.

Mr. Fieldner is internationally known as the inventor and developer of many techniques of testing and analyzing coal, coke and gas, and is recognized as having contributed to science a comprehensive knowledge of the properties and values of coal and related fuels. His pioneering work has resulted in advancing the standardization of methods of analysis of coals and other fuels. Because of the practical value of his original research, and because of his willingness to spread the benefits of his long experience, Mr. Fieldner is continually consulted by private industry and by public and Governmental organizations concerned with fuels.

In 1925, in connection with the ventilation of the Holland Vehicular Tunnel, investigations were carried on to determine the amount and composition of exhaust gases from motor vehicles under direction of Mr. Fieldner at the Pittsburgh Experiment Station.

Mr. Fieldner is active in many professional societies such as the *American Institute of Chemical Engineering,* the *American Chemical Society,* the *American Society for Testing Materials,* of which he was president in 1937, the *American Gas Association,* and the *American Institute of Mining and Metallurgical Engineers.* He has been closely identified with ASHVE research for many years and serves as ex-officio member of the Society's Committee on Research.

As chief of the Technologic Branch of the Bureau of Mines, he now directs such diversified activities as petroleum investigation, exploration for strategic minerals, metallurgical research on the ores of vital war metals and research work in the testing of coal, all of which currently is devoted to the successful prosecution of the War.

Graphical Symbols for Drawings

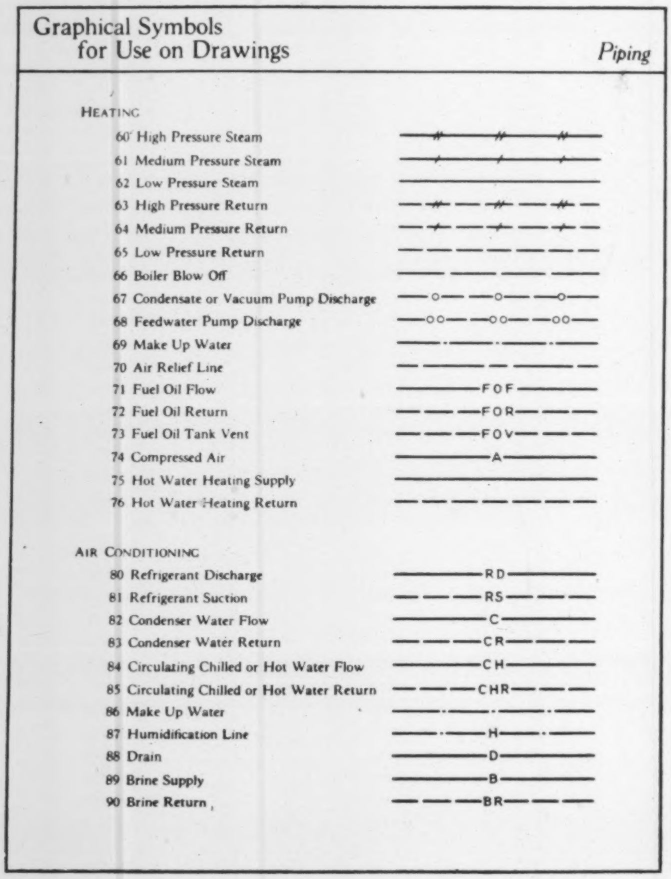
In November 1941 the *American Standards Association* approved a group of heating, ventilation, refrigeration, air conditioning, plumbing and welding graphical symbols for use on drawings which were designated as Z32.2-1941.

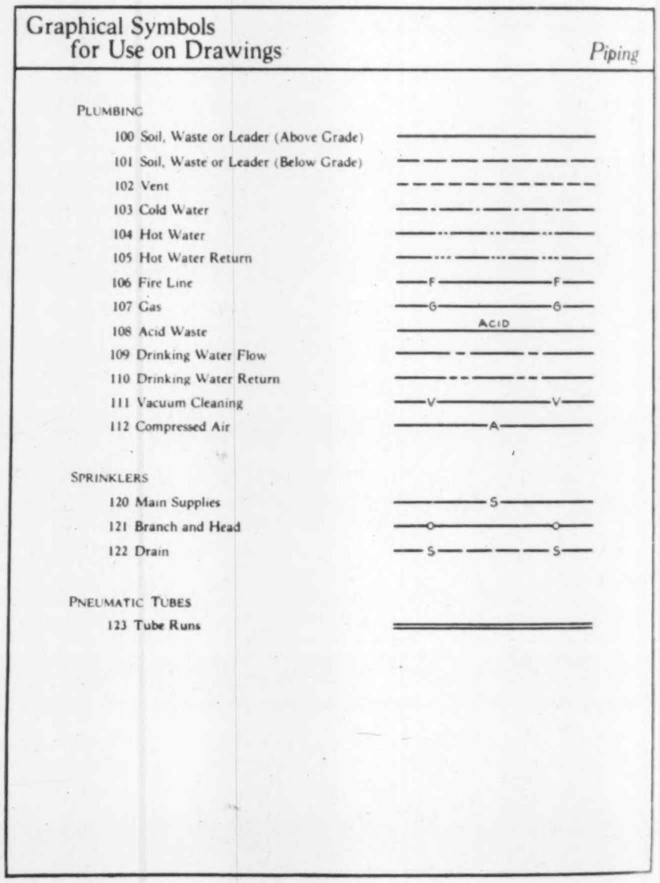
As early as December 1928, a committee was appointed to review a number of suggested series of symbols including (1) those proposed by the American Society of Heating and Ventilating Engineers in 1925 and (2) those proposed in 1926 by the *National Association of Master Plumbers of the United States.* The committee also drew on material from the publications of various organizations and in November 1935 the first American Standard was approved with a serial number of Z14.2-1935.

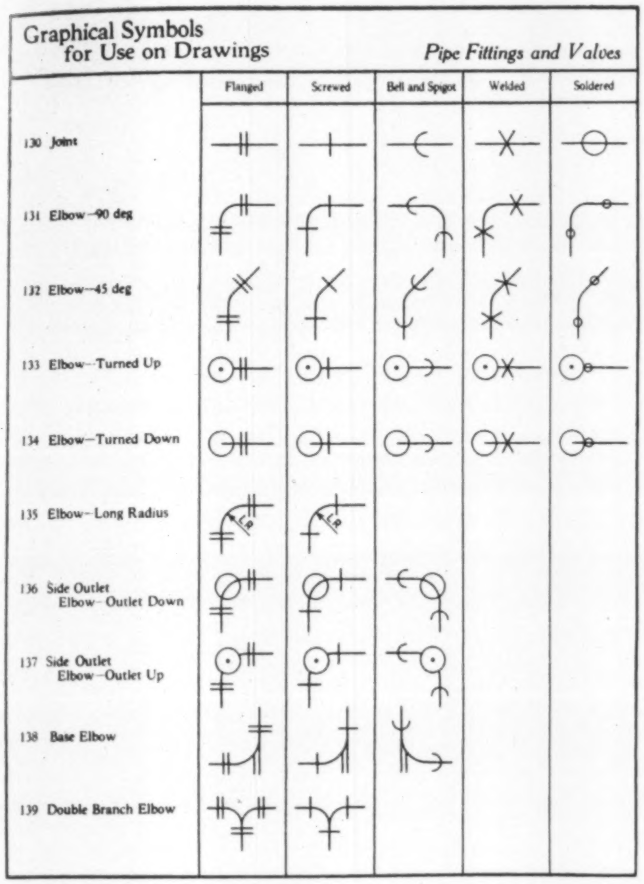
During the last stages of this activity a number of interested groups requested that the project be broadened. Accordingly, in November 1933, the *American Standards Association* called a conference to which were invited representatives of the Sectional Committees on Standardization of Letter Symbols and Abbreviations for Science and Engineering (Z10) and Standards for Drawings and Drafting Room Practice (Z14). As a result of this discussion it was agreed that all of the activity on graphical symbols should be concentrated in a new Sectional Committee on Standardization of Graphical Symbols and Abbreviations for Use on Drawings (Z32).

At the organization meeting of the new committee held in June 1936, the work was divided into two sub-committees, one covering symbols in Mechanical Engineering, and the other dealing with Electrical Engineering. To bring the existing standard in conformity with current practice, a revision of Z14.2-1939 was initiated by the sub-committee on Mechanical Engineering and its sub-groups, providing for the addition of symbols on air conditioning, sprinklers, pneumatic tubes, duct work, refrigerating and welding.

The sub-group responsible for revising and preparing the symbols for heating, ventilation, refrigeration and air conditioning, consisted of E. E. Ashley, ASHVE, New York, N. Y., *chairman;* L. L. Munier, *HPACCNA,* New York, N. Y., J. L. Murphy, *NAMP,* New York, N. Y., E. L. Penfrase, *ASSE,* New York, N. Y., L. H. Roller, *ASRE,* New York, N. Y., and E. R. Wolfert, *ACMA,* West Springfield, Mass.


Graphical Symbols for Use on Drawings
Piping
Heating
60 High Pressure Steam
61 Medium Pressure Steam
62 Low Pressure Steam
63 High Pressure Return
64 Medium Pressure Return
65 Low Pressure Return
66 Boiler Blow Off
67 Condensate or Vacuum Pump Discharge
68 Feedwater Pump Discharge
69 Make Up Water
70 Air Relief Line
71 Fuel Oil Flow
FOF
72 Fuel Oil Return
FOR
73 Fuel Oil Tank Vent
FOV
74 Compressed Air
A
75 Hot Water Heating Supply
76 Hot Water Heating Return
Air Conditioning
80 Refrigerant Discharge
RD
81 Refrigerant Suction
RS
82 Condenser Water Flow
C
83 Condenser Water Return
CR
84 Circulating Chilled or Hot Water Flow
CH
85 Circulating Chilled or Hot Water Return
CHR
86 Make Up Water
87 Humidification Line
H
88 Drain
D
89 Brine Supply
B
90 Brine Return
BR


Graphical Symbols for Use on Drawings
Piping
Plumbing
100 Soil, Waste or Leader (Above Grade)
101 Soil, Waste or Leader (Below Grade)
102 Vent
103 Cold Water
104 Hot Water
105 Hot Water Return
106 Fire Line
F
107 Gas
G
108 Acid Waste
Acid
109 Drinking Water Flow
110 Drinking Water Return
111 Vacuum Cleaning
V
112 Compressed Air
A
Sprinklers
120 Main Supplies
S
121 Branch and Head
122 Drain
S
Pneumatic Tubes
123 Tube Runs

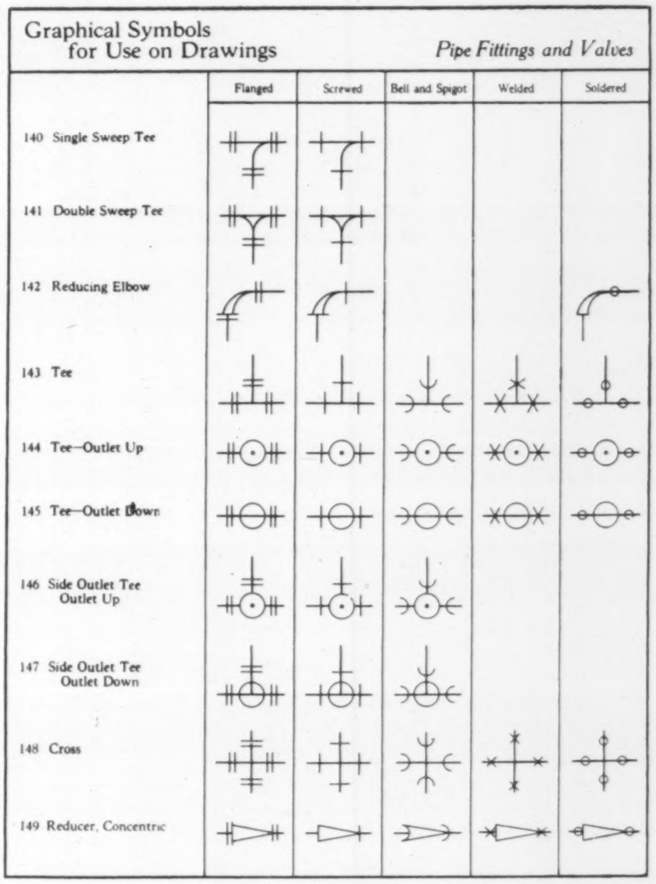


Graphical Symbols for Use on Drawings

Pipe Fittings and Valves

Flanged | Screwed | Bell and Spigot | Welded | Soldered

130 Joint
131 Elbow—90 deg
132 Elbow—45 deg
133 Elbow—Turned Up
134 Elbow—Turned Down
135 Elbow—Long Radius
136 Side Outlet Elbow—Outlet Down
137 Side Outlet Elbow—Outlet Up
138 Base Elbow
139 Double Branch Elbow

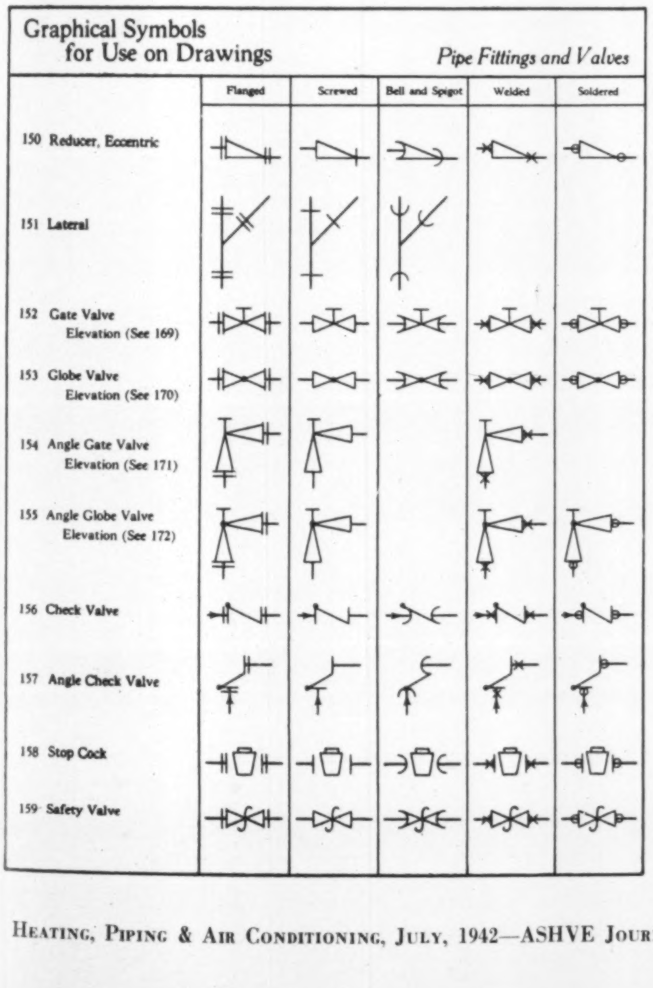


Graphical Symbols for Use on Drawings

Pipe Fittings and Valves

Flanged | Screwed | Bell and Spigot | Welded | Soldered

140 Single Sweep Tee
141 Double Sweep Tee
142 Reducing Elbow
143 Tee
144 Tee—Outlet Up
145 Tee—Outlet Down
146 Side Outlet Tee Outlet Up
147 Side Outlet Tee Outlet Down
148 Cross
149 Reducer, Concentric

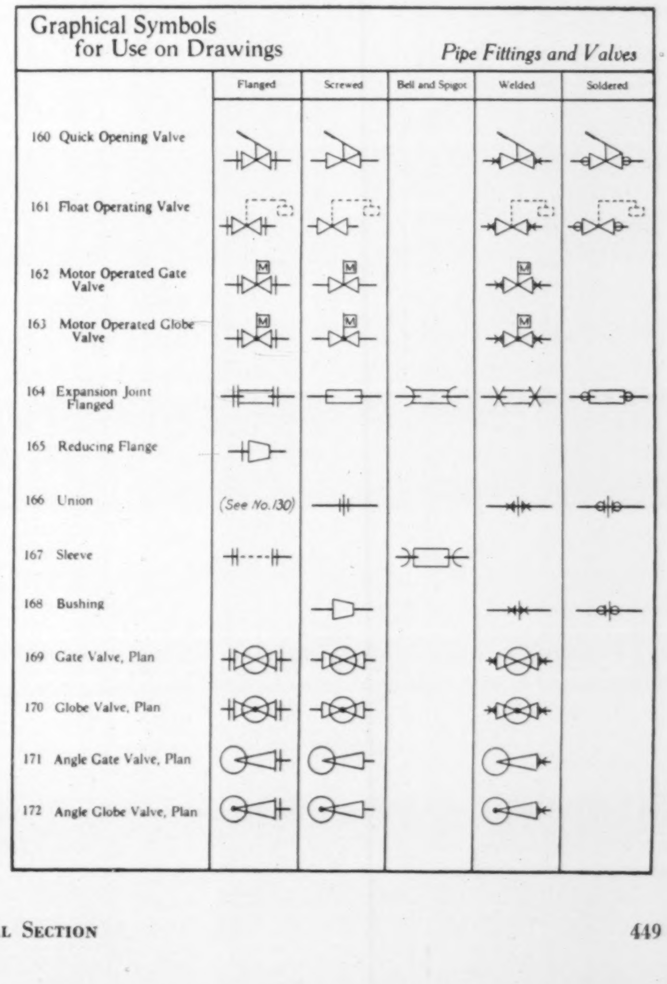


Graphical Symbols for Use on Drawings

Pipe Fittings and Valves

Flanged | Screwed | Bell and Spigot | Welded | Soldered

150 Reducer, Eccentric
151 Lateral
152 Gate Valve Elevation (See 169)
153 Globe Valve Elevation (See 170)
154 Angle Gate Valve Elevation (See 171)
155 Angle Globe Valve Elevation (See 172)
156 Check Valve
157 Angle Check Valve
158 Stop Cock
159 Safety Valve



Graphical Symbols for Use on Drawings

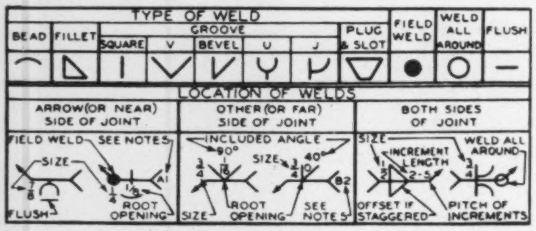
Pipe Fittings and Valves

Flanged | Screwed | Bell and Spigot | Welded | Soldered

160 Quick Opening Valve
161 Float Operating Valve
162 Motor Operated Gate Valve
163 Motor Operated Globe Valve
164 Expansion Joint Flanged
165 Reducing Flange
166 Union — Flanged: *(See No. 130)*
167 Sleeve
168 Bushing
169 Gate Valve, Plan
170 Globe Valve, Plan
171 Angle Gate Valve, Plan
172 Angle Globe Valve, Plan

Graphical Symbols for Use on Drawings — *Welding*

ARC AND GAS WELDING


TYPE OF WELD
BEAD
FILLET
GROOVE
SQUARE
V
BEVEL
U
J
PLUG & SLOT
FIELD WELD
WELD ALL AROUND
FLUSH
LOCATION OF WELDS
ARROW (OR NEAR) SIDE OF JOINT
OTHER (OR FAR) SIDE OF JOINT
BOTH SIDES OF JOINT
FIELD WELD
SEE NOTES
SIZE
7/8
FLUSH
1/4
1/8
ROOT OPENING
A1
INCLUDED ANGLE
90°
3/4
1/16
SIZE
3/4
0
40°
B2
SIZE
ROOT OPENING
SEE NOTE 5
SIZE
INCREMENT LENGTH
1/2
2-5
WELD ALL AROUND
3/4
OFFSET IF STAGGERED
PITCH OF INCREMENTS

1 The side of the joint to which the arrow points is the arrow (or near) side

2 Both-sides welds of same type are of same size unless otherwise shown.

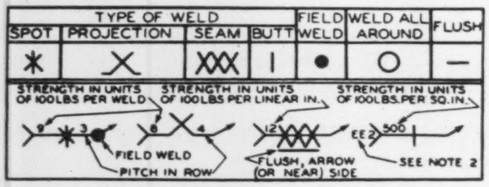
3 Symbols apply between abrupt changes in direction of joint or as dimensioned (except where all around symbol is used).

4 All welds are continuous and of user's standard proportions, unless otherwise shown

5 Tail of arrow used for specification reference (tail may be omitted when reference is not used)

6 Dimensions of weld sizes, increment lengths and spacings are given in inches

RESISTANCE WELDING


TYPE OF WELD
SPOT
PROJECTION
SEAM
BUTT
FIELD WELD
WELD ALL AROUND
FLUSH
STRENGTH IN UNITS OF 100LBS PER WELD
STRENGTH IN UNITS OF 100LBS PER LINEAR IN.
STRENGTH IN UNITS OF 100LBS.PER SQ.IN.
9
3
6
4
12
EE2
500
FIELD WELD
PITCH IN ROW
FLUSH, ARROW (OR NEAR) SIDE
SEE NOTE 2

1 Symbols apply between abrupt changes in direction of joint, or as dimensioned (except where all around symbol is used)

2 Tail or arrow used for specification reference (tail may be omitted when reference is not used)

3 All spacings are given in inches

NOTE: Further information on the use of these symbols will be found in "Welding Symbols and Instructions for Their Use," published by the American Welding Society, New York, N. Y., price 25 cents

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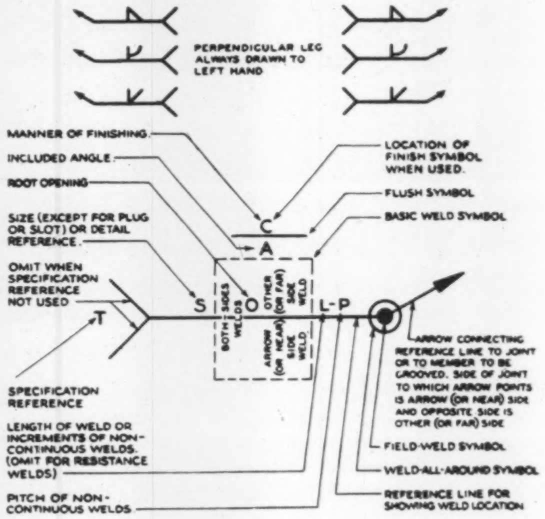
Graphical Symbols for Use on Drawings — *Welding*

LOCATION OF WELDING INFORMATION


PERPENDICULAR LEG ALWAYS DRAWN TO LEFT HAND
MANNER OF FINISHING.
INCLUDED ANGLE.
ROOT OPENING
SIZE (EXCEPT FOR PLUG OR SLOT) OR DETAIL REFERENCE.
OMIT WHEN SPECIFICATION REFERENCE NOT USED
SPECIFICATION REFERENCE
LENGTH OF WELD OR INCREMENTS OF NON-CONTINUOUS WELDS. (OMIT FOR RESISTANCE WELDS)
PITCH OF NON-CONTINUOUS WELDS
LOCATION OF FINISH SYMBOL WHEN USED.
FLUSH SYMBOL
BASIC WELD SYMBOL
C
A
S
O
T
L-P
BOTH SIDES WELDS
OTHER (OR FAR) SIDE WELD
ARROW (OR NEAR) SIDE WELD
ARROW CONNECTING REFERENCE LINE TO JOINT OR TO MEMBER TO BE GROOVED. SIDE OF JOINT TO WHICH ARROW POINTS IS ARROW (OR NEAR) SIDE AND OPPOSITE SIDE IS OTHER (OR FAR) SIDE
FIELD-WELD SYMBOL
WELD-ALL-AROUND SYMBOL
REFERENCE LINE FOR SHOWING WELD LOCATION

NOTE: Further information on the use of these symbols will be found in "Welding Symbols and Instructions for Their Use," published by the American Welding Society, New York, N. Y., price 25 cents

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Graphical Symbols for Use on Drawings — *Heating and Ventilating*

173 Heat Transfer Surface Plan
174 Wall Radiator Plan
175 Wall Radiator on Ceiling Plan
176 Unit Heater (Propeller) Plan
177 Unit Heater (Centrifugal Fan) Plan
178 Unit Ventilator Plan

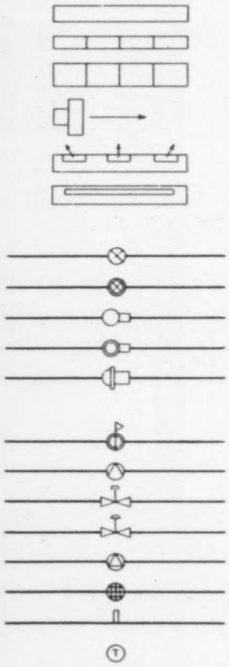
TRAPS

179 Thermostatic
180 Blast Thermostatic
181 Float and Thermostatic
182 Float
183 Boiler Return

VALVES

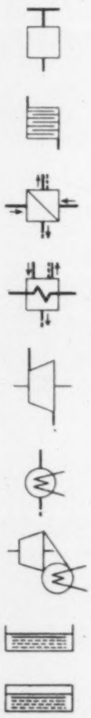
184 Reducing Pressure
185 Air Line
186 Lock and Shield
187 Diaphragm

188 Air Eliminator
189 Strainer
190 Thermometer
191 Thermostat

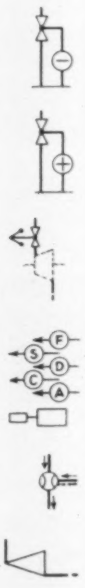


Graphical Symbols for Use on Drawings — *Heat-Power Apparatus*

240 Steam Generator (Boiler)
241 Flue Gas Reheater (Intermediate Superheater)
242 Live Steam Superheater
243 Feed Heater With Air Outlet
244 Steam Turbine
245 Surface Condenser
246 Condensing Turbine
247 Open Tank
248 Closed Tank



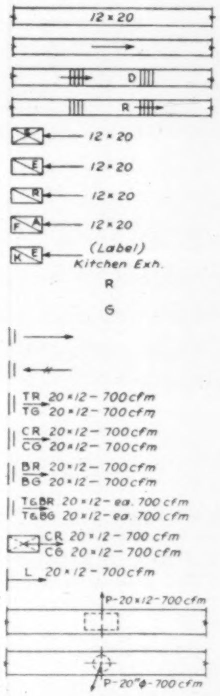
249 Automatic Reducing Valve
250 Automatic By-pass Valve
251 Automatic Valve Operated by Governor
252 Pumps
Boiler Feed
Service
Condensate
Circulating Water
Air
Reciprocating
253 Dynamic Pump (Air Ejector)
254 Steam Trap



Graphical Symbols for Use on Drawings

Ductwork

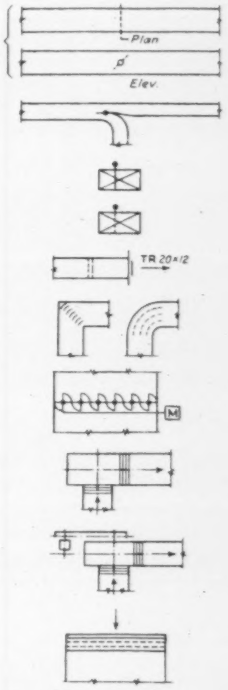
- 200 Duct (1st Figure, Width; 2nd, Depth)
- 201 Direction of Flow
- 202 Inclined Drop in Respect to Air Flow
- 203 Inclined Rise in Respect to Air Flow
- 204 Supply Duct Section
- 205 Exhaust Duct Section
- 206 Recirculation Duct Section
- 207 Fresh Air Duct Section
- 208 Other Ducts Section
- 209 Register
- 210 Grille
- 211 Supply Outlet
- 212 Exhaust Inlet
- 213 Top Register or Grille
- 214 Center Register or Grille
- 215 Bottom Register or Grille
- 216 Top and Bottom Register or Grille
- 217 Ceiling Register or Grille
- 218 Louver Opening
- 219 Adjustable Plaque


12 × 20
D
R
S
12 × 20
E
12 × 20
R
12 × 20
F
A
12 × 20
K
E
(Label)
Kitchen Exh.
R
G
TR 20 × 12 – 700 cfm
TG 20 × 12 – 700 cfm
CR 20 × 12 – 700 cfm
CG 20 × 12 – 700 cfm
BR 20 × 12 – 700 cfm
BG 20 × 12 – 700 cfm
T&BR 20 × 12 – ea. 700 cfm
T&BG 20 × 12 – ea. 700 cfm
CR 20 × 12 – 700 cfm
CG 20 × 12 – 700 cfm
L 20 × 12 – 700 cfm
P-20 × 12 – 700 cfm
P-20"ϕ-700 cfm

Graphical Symbols for Use on Drawings

Ductwork

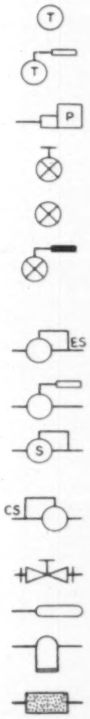
- 220 Volume Damper
- 221 Deflecting Damper
- 222 Deflecting Damper, Up
- 223 Deflecting Damper, Down
- 224 Adjustable Blank Off
- 225 Vanes
- 226 Automatic Dampers
- 227 Canvas Connections
- 228 Fan and Motor With Belt Guard
- 229 Intake Louvers and Screen


Plan
Elev.
TR 20 × 12
M

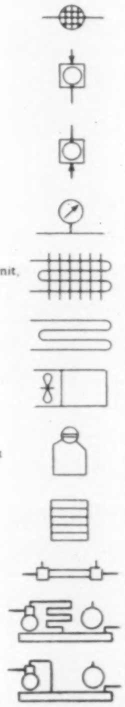
Graphical Symbols for Use on Drawings

Refrigerating

- 280 Thermostat (Self Contained)
- 281 Thermostat (Remote Bulb)
- 282 Pressure Switch
- 283 Hand Expansion Valve
- 284 Automatic Expansion Valve
- 285 Thermostatic Expansion Valve
- 286 Valve, Evaporator Pressure Regulating, Throttling Type (Evaporator Side)
- 287 Valve, Evaporator Pressure Regulating, Thermostatic Throttling Type
- 288 Valve, Evaporator Pressure Regulating, Snap-Action Valve
- 289 Valve, Compressor Suction Pressure Limiting, Throttling Type (Compressor Side)
- 290 Hand Shut Off Valve (Flanged)
- 291 Thermal Bulb
- 292 Scale Trap
- 293 Dryer



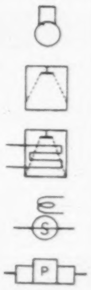
- 294 Strainer
- 295 High Side Float
- 296 Low Side Float
- 297 Gage
- 298 Finned Type Cooling Unit, Natural Convection
- 299 Pipe Coil
- 300 Forced Convection Cooling Unit
- 301 Immersion Cooling Unit
- 302 Ice Making Unit
- 303 Heat Interchanger
- 304 Condensing Unit, Air Cooled
- 305 Condensing Unit, Water Cooled



Graphical Symbols for Use on Drawings

Refrigerating

- 306 Compressor
- 307 Cooling Tower
- 308 Evaporative Condenser
- 309 Solenoid Valve
- 310 Pressure Switch With High Pressure Cut-Out



Graphical Symbols for Use on Drawings

Conventional Rivets

260 Shop Rivets, Two Full Heads

261 Shop Rivets, Countersunk and Chipped, Near Side

262 Shop Rivets, Countersunk and Chipped, Far Side

263 Shop Rivets, Countersunk and Chipped, Both Sides

264 Shop Rivets, Countersunk but Not Chipped, Max. ⅛ in. High Near Side

265 Shop Rivets, Countersunk but Not Chipped, Max. ⅛ in. High Far Side

266 Shop Rivets, Countersunk but Not Chipped, Max. ⅛ in. High Both Sides

267 Shop Rivets, Flattened to ¼ in. High for ½ in. and ⅝ in. Rivets Near Side

268 Shop Rivets, Flattened to ¼ in. High for ½ in. and ⅝ in. Rivets Far Side

269 Shop Rivets, Flattened to ¼ in. High for ½ in. and ⅝ in. Rivets Both Sides

270 Shop Rivets, Flattened to ⅜ in. High for ¾, ⅞, and 1 in. Rivets Near Side

271 Shop Rivets, Flattened to ⅜ in. High for ¾, ⅞, and 1 in. Rivets Far Side

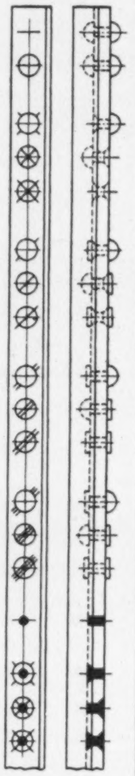
272 Shop Rivets, Flattened to ⅜ in. High for ¾, ⅞, and 1 in. Rivets Both Sides

273 Field Rivets, Two Full Heads

274 Field Rivets, Countersunk and Chipped, Near Side

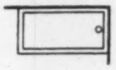
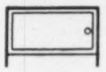
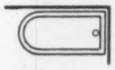
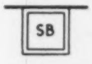
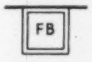
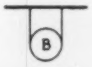
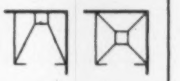
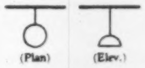
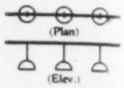
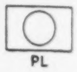
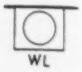
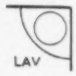
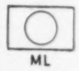
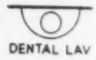
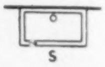
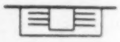
275 Field Rivets, Countersunk and Chipped, Far Side

276 Field Rivets, Countersunk and Chipped, Both Sides



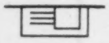
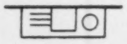
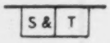
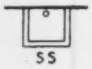
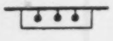
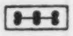
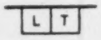
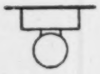
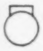
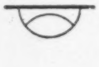
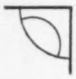
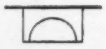
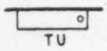
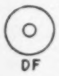
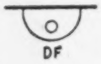
Graphical Symbols for Use on Drawings

Plumbing

No.	Name	Symbol
1	Corner Bath	
2	Recessed Bath	
3	Roll Rim Bath	
4	Sitz Bath	
5	Foot Bath	
6	Bidet	
7	Shower Stall	
8	Shower Head	
9	Overhead Gang Shower	
10	Pedestal Lavatory	
11	Wall Lavatory	
12	Corner Lavatory	
13	Manicure Lavatory Medical Lavatory	
14	Dental Lavatory	
15	Plain Kitchen Sink	
16	Kitchen Sink R & L Drain Board	

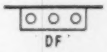
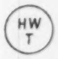
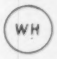
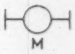
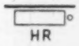
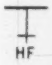
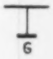
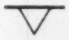
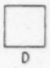
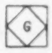
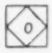
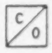
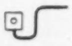
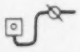
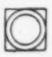
Graphical Symbols for Use on Drawings

Plumbing

No.	Name	Symbol
17	Kitchen Sink L. H. Drain Board	
18	Combination Sink and Dishwasher	
19	Combination Sink and Laundry Tray	
20	Service Sink	
21	Wash Sink (Wall Type)	
22	Wash Sink	
23	Laundry Tray	
24	Water Closet (Low Tank)	
25	Water Closet (No Tank)	
26	Urinal (Pedestal Type)	
27	Urinal (Wall Type)	
28	Urinal (Corner Type)	
29	Urinal (Stall Type)	
30	Urinal (Trough Type)	
31	Drinking Fountain (Pedestal Type)	
32	Drinking Fountain (Wall Type)	

Graphical Symbols for Use on Drawings

Plumbing

No.	Name	Symbol
33	Drinking Fountain (Trough Type)	
34	Hot Water Tank	
35	Water Heater	
36	Meter	
37	Hose Rack	
38	Hose Faucet	
39	Gas Outlet	
40	Vacuum Outlet	
41	Drain	
42	Grease Separator	
43	Oil Separator	
44	Cleanout	
45	Garage Drain	
46	Floor Drain With Backwater Valve	
47	Roof Sump	

Members on Active Duty*

U. S. ARMY

BACHOFER, Henry A., Jr., Private, 87th Air Base Squadron.

BADGETT, W. Howard, Capt., 360th Infantry School.

BARNES, Hugh S., First Lieut., 99th C.A. (aa).

BERNARD, Edgar L., Lieut., 23rd Armored Engr. Battalion.

BERZELIUS, Carl E., Capt., Battalion Comdr.

BROWN, Harper J., First Lieut., Ordnance Dept.

BRUNDAGE, F. Ward, First Lieut., 423rd C.A.

BURNS, Frank G., Capt., Infantry.

CAMPBELL, Alfred Q., Jr., Lieut., Field Artillery.

CAMPBELL, Bowen, Capt., B.O.C. No. 49.

CAMPBELL, George W., First Lieut., Air Corps.

CASKEY, Luther H., Jr., Second Lieut., Personnel Officer.

CHASE, R. E., Jr., Private First Class, 417th Ordnance Dept., A.V.N.

CHEESEMAN, Evans W., Lieut., Personnel Adjutant.

CLARK, Albert C., Capt., A.P.O. 843, Seattle, Wash.

CLEMENS, Joseph D., Second Lieut. (Engrs.) Air Corps.

CROLEY, Jack G., First Lieut., 76th Coast Artillery.

DABBS, John T., First Lieut., C.A.C.

DANIELSON, Wilmot A., Brig.-Gen. Q.M.C.

DICKSON, Robert W., Jr., Lieut., 40th Bombardment Group.

DRUM, Leo J., Jr., Second Lieut., Q.M.C.

ESCHENBACH, Sam P., First Lieut., 62nd C.A. (aa).

FELDSTEIN, Harold, First Lieut., Ordnance Dept.

FINERAN, Edward V., First Lieut., Signal Corps School.

FLARSHEIM, C. A., Private, E.R.C., Air Corps.

FORDERBRUGGEN, Kevin J., First Lieut., 65th C.A. (aa).

FOSTER, John G., Cadet, U. S. Air Corps.

FRANKLIN, S. H., Jr., Major, Ordnance Dept.

FREEMAN, Alfred W., Aviation Cadet, Squadron F.

FRIEDMAN, D. Harry, Capt., C.E., Asst. Utilities Officer.

GAULT, George W., Lieut., Corps of Engrs.

GONZALEZ, Rafael A., Capt., Ordnance Dept., U. S. Army.

GRABMAN, Henry B., Lieut., 36th Engineers (c).

GREEN, Everett W., Private, 87th Battalion.

HAWES, Harold D., Master Serg., O.C.S.

HERBERT, Richard M., Lieut., Air Corps.

HUST, Carl E., Capt., Ordnance Dept.

IBISON, James L., First Lieut., Asst. Constructing Quartermaster.

JONES, James T., Second Lieut., Engineer Reserve Corps.

KILLOREN, Donald E., Hdqs. Co., 15th Infantry.

KUMMER, Calvin J., First Lieut., 72nd F.A.

KURTZ, Robert W., First Lieut., Ordnance Works.

LENONE, J. M., Major, Corps of Engrs.

LIGHT, John C., First Lieut., Ordnance Dept.

LINEBAUGH, John E., Capt., Air Corps.

MACEACHIN, Grahm C., Capt., Corps of Engrs.

MACGREGOR, Cecil M., Capt., F.A.

MALIN, Benjamin S., First Lieut., Ordnance Dept.

MARSTON, Anson D., Major, G.S.C.

MCCAIN, H. King, Lieut., Quartermaster School

MCGINN, George F., Major, Infantry.

MCKAY, Albert W., Second Lieut., Ordnance Dept.

MORTON, Harold S., Major, Ordnance Dept.

NEAL, James P., Capt., Ordnance Dept.

NORRINGTON, Walter L., First Lieut., Ordnance Dept.

PARSONS, Leonard D., Jr., Capt., Q.M.C.

PELLMOUNTER, T. V., Second Lieut., 55th Ordnance Co.

PRAWL, Frank E., Lieut., Ordnance Dept.

RHINE, George R., Capt., Ordnance Dept.

RUEMMELE, Albert M., First Lieut., Ordnance Section PRGD.

SANDFORT, John F., First Lieut., F.A., B.O.C. No. 44.

SAURWEIN, G. K., Major, Ordnance, U. S. Army.

SCHLICK, Paul F., Colonel, 216th Coast Artillery.

SCOTT, Allison F. H., Lieut. Colonel, Coast Artillery Corps.

SHARP, John R., Major, Corps of Engrs.

SHEARER, William A., Jr., Second Lieut., Engr. Corps.

SLOANE, David J., 56th General Hospital.

SNYDER, Edwin F., Jr., Second Lieut., Corps of Engrs.

SOBEL, Frank, Private, Co. A, 101 M.P. Bn.

SPENCE, Robert A., Lieut., Ordnance Dept.

STERNER, Douglas S., First Lieut., 282nd Quartermaster Corps.

VANNOUHUYS, Herbert C., First Lieut.

WEAVER, J. V. O., Major, Air Corps.

WILSON, Westray E., Major, Q.M.C.

ZINK, David D., Major, G.S.C., 1st Arm'd Corps.

U. S. NAVY

AKERS, George W., Lieut. Comdr., U.S.S. Markab.

BABER, John E., Ensign, U. S. Navy Recruiting Station.

BAHLMANN, W. F., Lieut., U. S. N. R.

BAKER, Roland H., Lieut. Comdr., U. S. N. R.

CARY, Edward B., Lieut. Comdr. (CEC), U. S. N. R.

COVER, R. R., Warrant Officer, EV(s), U. S. N. R.

HERRE, Harold A., Ensign, E.V.S., U. S. N. R.

HOUGHTEN, F. C., Lieut. Comdr., U. S. N. R.

KAHN, Charles R., Jr., Ensign, Philadelphia Navy Yard.

KELLY, Olin A., Aviation Cadet, U. S. Army Air Force.

LADD, David, Lieut. Comdr. U. S. N. R.

LANDAUER, Leo L., Lieut., U. S. N. R.

MABLEY, Louis C., Lieut. Comdr., U. S. N. R.

MATHEKA, Charles R., U.S.S. Tattnall.

MATTHIES, Leo A., Ensign, U. S. N. R.

MILLARD, Junius W., Lieut. (s.g.) Organized Reserve.

MULCEY, Paul A., Lieut. (s.g.) U. S. N. R.

PARKINSON, John S., Lieut., U. S. N. R.

POWERS, Edgar C., Lieut. (jg) U. S. N. R.

QUEER, E. R., Lieut. (E-V) (S) U. S. N. R.

RODEE, E. J., Lieut., U. S. N. R.

SHAPIRO, Charles A., Ensign, D-V (s) U. S. N. R.

SIMPSON, Robert L., Ensign (E-V) (s) U. S. N. R.

STACEY, Alfred E., Jr., Comdr., U. S. N. R.

STEEL, R. Justin, Lieut. (jg) U. S. N. R.

STERNE, C. M., Lieut. Comdr., U. S. N. R.

URDAHL, Thomas H., Lieut. Comdr. (E-V) (s)

WATT, Richard M., Jr., Comdr.

WESTPHAL, N. E., Ensign.

FOREIGN MEMBERS

ARMOUR, Edson G., Royal Canadian Air Force.

BALLANTYNE, George L., Instructor, Royal Canadian Air Force.

BARNETT, Campbell M., Canadian Active Service Forces.

BISHOP, Joseph W., Canadian Active Service Forces.

BODMER, Emmanuel, Lieut., Air Force.

BOWERMAN, E. L., Flying Officer, Royal Canadian Air Force.

BUTT, Roderick E. W.

DANIEL, William E., Second Lieut., Royal Artillery.

DOWLER, Edward A., First Lieut., Royal Canadian Engineers.

FOSTER, Philip H., Pilot Officer, Royal Canadian Air Force.

FOX, John H., Canadian Active Service Forces.

JOHNS, Charles F., Flight Lieut., Royal Canadian Air Force.

KITCHEN, William H., Sub-Lieut., Royal Canadian Naval Volunteer Reserves.

LABONNE, Henri, Lieut., "Les Fusiliers de Sherbrooke."

MACLACHLAN, Victor D., Flight Lieut., Royal Air Force Volunteer Reserves.

NORFOLK, Leslie W., Major, Royal Engrs.

PENNOCK, William B., Major, Royal Canadian Engrs.

POUGHER, Bernard R. E., Sergt., 110(el) A. T. Coy R. E.

PRICE, Ernest H., Lieut., Royal Canadian Engrs.

PRYKE, John K. M., Capt., Royal Army Ordnance Corp.

RICHARDSON, Robert D., Bombardier, Royal Regiment Artillery.

ROBINSON, Jack A.

SMITH, Gerald E., Royal Canadian Engrs.

WILKINSON, Arthur, Flying Officer, Royal Canadian Air Force.

WOOD, Alfred W., Royal Canadian Air Force.

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*Compiled as of July 1, 1942.

NEW OFFICERS INDUCTED AT WASHINGTON, D. C.

May 13, 1942. The final meeting of the season was held by the Washington, D. C., Chapter at the Dodge Hotel, and inasmuch as it was previously announced that this would be a business meeting only, there were but 9 members and guests present.

After dinner the group assembled in the Garden House, and Pres. F. A. Leser called the meeting to order. The previous minutes were read by the secretary and approved as read. K. F. Baldwin reported as treasurer, followed by a report on the membership activities by J. H. Fogg, reporting that the Chapter had increased approximately 14 per cent in membership.

F. B. Sale presented a resolution, recommending that the Chapter prepare a suitable scroll on which the names of all Chapter members in the armed services be inscribed. The resolution was unanimously adopted, and Mr. Sale also advocated the inclusion of telephone numbers of the members along with their addresses in the Kardex filing system. It was agreed that the Secretary make the necessary addition.

Lt. Comdr. F. C. Houghten, U.S.N.R., director of the Society's Research Laboratory at Pittsburgh, was introduced, and told the members that he hoped to be in regular attendance at the Washington Chapter meetings in the future.

Proceeding to the business of the evening, President Leser appointed Messrs. Van Alsburg, Littleford and Baldwin, as tellers to count the votes in the annual election of officers, and they reported as follows:

President—R. S. Dill
Vice-President—F. M. Thuney
Secretary—J. W. Markert
Treasurer—F. B. Sale
Board of Governors—F. A. Leser, S. L. Gregg, W. H. Littleford.

The officers for the 1942-1943 season were immediately inducted into office. Mr. Leser, retiring president, expressed thanks for the cooperation of the members during the past year and praised the work of his committee members and fellow officers.

With no further business, President Dill declared the meeting adjourned at 9:30 p.m.

ASHVE MEMBERS ADDRESS SPEE; WOODS ELECTED SECOND VICE-PRESIDENT

At the three-day meeting of the *Society for the Promotion of Engineering Education* held at Columbia University, New York, June 27-29, W. L. Fleisher, Past President of the Society, addressed the mechanical engineering section on "Physiological Aspects of Heating and Air Conditioning as it Should be Taught to Engineering Students."

John James, Technical Secretary of the Society, discussed the Heating and Air Conditioning Curriculum, and G. L. Tuve, Professor of heat power engineering, Case School of Applied Science, Cleveland, presented a discussion on Heating and Ventilating.

B. M. Woods, Professor of mechanical engineering, University of California, Berkeley, and a member of the Council of the Society, and R. B. Rice, Professor of experimental engineering, North Carolina State College, Raleigh, registered at the opening session. Professor Woods was elected second vice-president of the *SPEE.*

KANSAS CITY MEMBERS MEET

May 11, 1942. The monthly meeting of the Kansas City Chapter was held at the President Hotel, with 12 members present for the buffet supper preceding the meeting. Pres. Gustav Nottberg opened the meeting, and the minutes of the previous meeting were approved as read.

L. E. Boyd submitted the treasurer's report, followed by a letter from Prof. E. O. Eastwood, president of the ASHVE, expressing his appreciation for flowers and attention of Chapter members during his illness. A letter from C. F. Boester concerning a program to obtain special funds for research activities was also read.

President Nottberg then appointed J. M. Arthur, Jr., to serve as chairman of the nominating committee, with Carl Clegg and M. J. Stevenson as committee members. A motion was made, properly seconded and carried, to change the date of the annual spring party from June to the first regular fall meeting, and to have the June meeting a regular monthly meeting.

ELECTION OF OFFICERS AT IOWA

May 12, 1942. The regular monthly meeting of the Iowa Chapter was held at the Chamberlain Hotel, Des Moines, with Pres. Perry La Rue presiding. The minutes of the previous meeting were read and approved, and President La Rue appointed the tellers and the election resulted in the following:

President—F. E. Triggs
Vice-President—C. W. Helstrom
Secretary-Treasurer—W. W. Stuart
Board of Governors—E. H. Borg, L. E. Hedeen.

A report for the ASHVE on contributions and research was read and referred to the newly elected officers.

On motion of Prof. R. A. Norman and seconded by A. L. Walters, and carried, it was voted to exempt all members of the Iowa Chapter, who are in any of the armed services of the United States, from payment of Iowa Chapter dues.

A report by J. F. Collins, Jr., Pittsburgh, chairman of the Chapter Relations Committee, on local chapter dues and the handling of new members was read and submitted to the newly elected officers.

April 14, 1942. The April meeting of the Iowa Chapter was held at the Iowa Memorial Union, Iowa City, and President La Rue also presided. Following the reading of the minutes of the March meeting, which were approved as read, the nominating committee submitted its report.

An interesting program was arranged by Prof. H. O. Croft, consisting of a very instructive two hour visit to the Hydraulics Laboratory at the State University of Iowa, Ames. An interesting exhibit was a miniature scale of the locks for the Sault Ste. Marie Canal. A demonstration of the hydraulics of plumbing was thoroughly enjoyed, especially as it applied to back siphonage and cross connection.

After dinner, Dr. Hunter Rouse, professor of fluid mechanics, presented a very interesting talk on the flow of liquids, using slides to illustrate the demonstration, which presented a very clear picture of laminar and turbulent flow.

A vote of appreciation was extended to Professor Croft for the very excellent program he arranged, followed by adjournment.

MINNESOTA HOLDS ELECTION

May 11, 1942. The annual golf and dinner party was held at the Midland Hills Country Club, Minneapolis, with 45 members and guests in attendance.

The nominating committee submitted their report and the following officers were elected for the year 1942-43:

President—William McNamara
Vice-President—D. B. Anderson
Secretary-Treasurer—A. W. Schultz
Board of Governors—J. F. Stafford, F. H. Schernbeck.

The officers elected took office immediately, followed by a discussion of the plans for the Semi-Annual Meeting in St. Paul, by N. D. Adams, M. S. Wunderlich and Mr. Stafford.

All who attended this get-together claimed it to be one of the most successful held by the Minnesota Chapter. Mr. Schernbeck was in complete charge of arrangements.

NEW OFFICERS INSTALLED AT ILLINOIS

May 11, 1942. The 36th annual meeting of the Illinois Chapter was held at the Merchants and Manufacturers Club in the Merchandise Mart, Chicago, with 110 members, ladies and guests in attendance, which was the best attendance of any annual meeting.

After dinner Pres. I. E. Brooke called the meeting to order and spoke briefly on the Chapter's place in the war effort. Secretary M. W. Bishop then read the Board of Governors' report on the year's activities. A. O. May, treasurer, presented the report of the finance committee, and J. J. Hayes reported that the auditing committee had found the report to be correct.

The report of the meetings and publications committee was given by C. M. Burnam, Jr., chairman of the committee, and J. H. Milliken, chairman of the board of tellers, reported on the annual election and declared the following officers elected for the coming year:

President—E. M. Mittendorff
Vice-President—M. W. Bishop
Secretary—A. O. May
Treasurer—W. A. Kuechenberg
Board of Governors—C. M. Burnam, Jr., W. A. Stahl, K. C. Porter.

The retiring president presented President Mittendorff with a new gavel and turned the meeting over to him. President Mittendorff expressed his gratitude and pledged himself to give his best efforts to the Chapter. After introducing the newly elected officers, President Mittendorff introduced John Howatt, past president of the Society and chief engineer of the Chicago Board of Education, who spoke on the Chicago Civilian Defense Plan for the City's schools. Mr. Howatt is in charge of this phase of Civilian Defense in Chicago and his description of the arrangements made for the protection of school children in the event of air raids impressed the audience with its thoroughness and detail.

Mr. Howatt then introduced Anthony J. Mullany, Chief Fire Marshal of the City of Chicago, who told of the plans and organization of the Fire Department for use in an emergency and pointed out that Chicago's inland location was not definite assurance that it could not be raided, since Chicago is actually 25 miles closer to the nearest German controlled air base than is New York City. He requested above all that if bomb attacks do come to remain calm, help avert panic and cooperate with the Fire and Police Departments to the fullest extent.

Lt. Harry Wolf of the Chicago Fire Department discussed the various types of incendiary bombs, explaining how they operate and pointing out that on long-distance raids much greater and widespread damage is possible by a given number of enemy bombers carrying a large number of two-pound incendiary bombs than would result from the smaller number of much heavier demolition bombs that the same number of planes could carry.

Sound moving pictures of the *Bombing Attack on London in December 1940* and *How to Handle Incendiary Bombs* were shown. However, he pointed out that the apparatus used in the second picture was not available to the ordinary apartment dweller, and called attention to the fact that the Chicago Fire Department has conducted a series of exhaustive tests in the handling of incendiary bombs with implements readily available to all. These tests were made into a color moving picture and the Chapter was privileged to witness the first public showing of this film. Lieutenant Wolf emphasized the necessity for the individual civilian to be able to handle incendiary bombs, by pointing out that during a raid the Fire Department would be too busy with large fires to answer each individual call for aid. Lieutenant Wolf's concise instructions on handling this type of bomb left even the ladies feeling that incendiary bombs are not so bad when you get to know them. Following a series of questions and answers, sound pictures of the bombing of Pearl Harbor and the burning of the Normandie were shown.

Chief Mullany closed the meeting with a plea to everyone to avoid carelessness, which every year causes a staggering fire loss. The meeting adjourned at 10:50 p.m.

BLACKOUT COMMITTEE REPORTS AT DELTA MEETING

May 12, 1942. The regular monthly luncheon meeting of the Delta Chapter was held at Arnaud's Restaurant, with 7 members and guests in attendance.

The meeting was called to order by Pres. G. C. Kerr, followed by a report on the progress of the Blackout Committee, by J. J. Friedler and G. A. Hero, Jr. This committee is comprised of 6 members from the Delta Chapter, three members are regular members and three are what is termed *floating members.* It is expected at the next meeting the Committee will give its final report which will be submitted to the local Civilian Defense office.

NORTH TEXAS HOLDS MEETINGS

May 11, 1942. The regular monthly meeting of the North Texas Chapter was held at the Stoneleigh Hotel, Dallas, with 12 members and guests attending. The meeting was called to order by Pres. T. H. Anspacher, and the minutes of the previous meeting were approved as read.

President Anspacher then turned the meeting over to D. C. Pfeiffer, whose subject was *Drying of Human Blood Plasma for the Army and Navy.* Mr. Pfeiffer helped to a very great extent to perfect the system now used by Baylor Institute in the drying of blood plasma and has been consulted generally by others interested in this work. His talk was interesting as well as instructive, telling of the system of collecting blood samples by the Red Cross, 25 samples of the various types being pooled and neutralized, 1 sample out of each 25 being tested for purity or sterility. The cells of the blood, he explained, are then separated from the plasma by a centrifuge, the plasma is then prefrozen or shell frozen and dried in special bottles which are afterwards incorporated in field kits by the Army or Navy to be used when needed.

Mr. Pfeiffer also stressed the fact that the administering of blood plasma does not take the place of a blood transfusion, but merely brings the patient out of shock, or primes the pump, which in this case is the human heart.

A vote of thanks was extended Mr. Pfeiffer after an extended period of discussion. Following a report by the program and membership committees the meeting adjourned.

April 13, 1942. At the regular monthly meeting held at the Stoneleigh Hotel, 18 members and guests were present for dinner and the meeting. The minutes of the March meeting were read and approved, followed by a report of the finances of the Chapter, as well as reports from the program and membership committees.

President Anspacher introduced Mr. Richardson of the Allis-Chalmers Co., who discussed the sound film *We Work for Victory and Plan for Peace.*

GOLDEN GATE ELECTS OFFICERS

May 6, 1942. In the absence of Pres. C. E. Bentley, the meeting was called to order by Vice-Pres. J. A. Hill and 24 members and guests were present.

As this was the annual meeting of the chapter Mr. Hill reviewed the activities during the past year and complimented President Bentley on the fine work done during his term of office, and as spokesman for President Bentley, he thanked the officers and members for the splendid cooperation given during the year.

Complete reports were submitted and read by the various committee chairmen, and these were all approved as read.

Mr. Hill asked for a vote of thanks to E. W. Simons for the work he did with his committee on technical matters, and expressed the hope that the efforts of the Golden Gate Chapter along these lines would be continued.

Mr. Hill then called on E. H. Goins, chairman of the tellers committee to make his report on the ballot. He reported that the vote was unanimous for the officers nominated and the following were elected for the coming season:

President—J. F. Kooistra
Vice-President—R. A. Folsom
Secretary—C. L. Peterson
Treasurer—R. B. Holland
Board of Governors—C. A. Bentley, Mason Emanuels, James Gayner, O. W. Johnson.

After the report of the tellers Mr. Hill presented the gavel and the Chapter Charter to the incoming president and turned the meeting over to the newly elected officers.

With no further business to come, Mr. Hill, as chairman of the entertainment committee, presented a splendid film titled *Airacobra,* portraying the manufacture and operation of the high-speed and powerful pursuit planes manufactured by the Bell Corp.

WESTERN NEW YORK ELECTS OFFICERS

May 11, 1942. The May meeting of the Western New York Chapter was held at the University Club, Buffalo, with Pres. W. R. Heath presiding.

B. C. Candee submitted the treasurer's report, and President Heath then called attention to a letter received from C. F. Boester, St. Louis, chairman of Finance of the Committee on Research, asking that a committee to solicit funds for a research building be appointed. The secretary was instructed to communicate with the Secretary of the Society to secure more complete details and information concerning Mr. Boester's request, and the matter was then turned over to the Board of Governors for action.

The president then appointed the tellers to count the ballots which were filed by mail. Edwin Woolcock reporting for the tellers announced the following as the newly elected officers:

President—H. C. Shafer
1st Vice-President—S. M. Quackenbush
2nd Vice-President—S. W. Strouse
Secretary—Herman Seelbach, Jr.
Treasurer—B. C. Candee
Board of Governors—M. C. Beman, Joseph Davis, Roswell Farnham, W. R. Heath, B. J. Mahoney.

The incoming president then took over the gavel and called upon L. P. Saunders to present the past president's certificate to Retiring President Heath.

It was then moved and seconded that arrangements be made for a joint golf meet between the ASHVE and the *Air Conditioning Council.* President Shafer appointed a committee under the chairmanship of Mr. Strouse, with Messrs. Mahoney and Seelbach as members to arrange for this golf meet.

The speaker of the evening was then introduced, the Right Honorable Geoffrey Smyth-Wells, K.C.B., a member of the British Procurement Division, who was passing through Buffalo on his way from Toronto to Washington. He gave a most interesting discussion which he delivered in a very humorous style. His talk was well received and it came as no great surprise after he had finished that instead of the Honorable gentleman he was introduced as, he turned out to be one John Summers with a real flare for entertainment and a native of Buffalo with a broad British accent.

As reported by the secretary, this was one of the most amusing speeches the Western New York Chapter had ever listened to.

OREGON MEMBERS MEET

April 9, 1942. The Oregon members met at the Mallory Hotel, Portland, with Vice-Pres. B. W. Farnes presiding in the absence of Pres. J. F. McIndoe.

The annual report of the Chapter's research committee, submitted by J. Donald Kroeker, chairman, stated that apparatus for the study of heat transmission through wet walls is approaching completion.

F. A. Olmsted, technical director of the Crown Willimette Paper Co., Camas, Wash., was the speaker of the evening. He described the manufacturing methods employed in the production of pulp and paper, and traced the course of the wood through the mill from the log boom to the converted paper product. Following his talk, Mr. Olmsted showed illustrative motion pictures taken at the Camas plant.

The newly elected officers were then installed as follows:

President—B. W. Farnes
Vice-President—J. A. Freeman
Treasurer—F. F. Urban
Secretary—R. E. Chase, Jr.
Board of Governors—W. R. Burton, J. F. McIndoe, W. J. Widmer.

A request from the City of Portland for the Chapter's advice in the technical aspects of a proposed fuel conservation program, directed toward alleviation of the threatened shortage of liquid and solid fuels, was read and discussed. The Chapter selected a committee to discuss the program with the city commissioners.

WESTERN MICHIGAN HEARS M. F. BLANKIN

April 14, 1942. The April meeting of the Western Michigan Chapter was held at the Union Building, Michigan State College, East Lansing, and 45 members and guests were present. The meeting was called to order by Pres. W. G. Schlichting, and following the reading of the minutes of the previous meeting, he called for a report from the chairman of each of the committees functioning. Messrs. H. D. Bratt, T. D. Stafford and W. W. Bradfield responded for their respective committees.

The following names were submitted by the nominating committee: *President*—F. C. Warren; *Vice-President*—C. H. Pesterfield; *Secretary*—H. D. Bratt; *Treasurer*—V. H. Hill; *Board of Governors*—W. G. Schlichting, W. D. DeRoo, H. J. Metzger.

President Schlichting called for a discussion of a proposed meeting schedule for every other month next year, due to the gasoline situation and rubber shortage.

W. G. Boales then gave a short talk on the Rackham Memorial Building in Detroit.

Prof. L. G. Miller then introduced the guest of the evening, M. F. Blankin, Philadelphia, Pa., First Vice-President of the Society, who gave a very interesting talk on present day practices in hot water heating, which the members enjoyed, followed by an informal discussion.

The secretary was then instructed to mail ballots to each member for election of officers for 1942-1943 season, followed by adjournment.

MANITOBA ANNOUNCES NEW OFFICERS

April 15, 1942. The April meeting of the Manitoba Chapter was called to order by Pres. R. L. Kent, at the Fort Garry Hotel, with 13 members present.

The minutes of the special meeting held on March 18, were approved as read, and President Kent read several letters relative to Society affairs. William Glass, chairman of the membership committee, reported the possibility of securing several new members.

President Kent advised the members of J. B. Steele's illness, and reported that Mr. Steele was making good progress and was expected to leave the hospital shortly. The meeting unanimously agreed to send a telegram to Mr. Steele.

William Glass then reported on the Annual Meeting proceedings in Philadelphia, and concluded by pointing out that it seemed quite possible that unless immediate action was taken there would not be any Canadian representative on the Council. After considerable discussion it was proposed by D. F. Michie that the president of the Chapter communicate with the Presidents of the Ontario and Montreal Chapters, stating this fact and urging that action be taken toward possible Canadian nominations for the Council.

The nominating committee reporting for the Manitoba Chapter announced the following as the newly elected officers for the year 1942-43:

President—Ivan McDonald
Vice-President—F. L. Chester
Secretary-Treasurer—Einar Anderson
Board of Governors—R. L. Kent, E. M. Hepburn, F. T. Ball.

Mr. Michie then addressed the meeting and led a general discussion on several items of interest, and it was agreed that the discussions on the various papers published in the Society's Journal are beneficial to all.

J. I. LYLE, CARRIER PRESIDENT, DIES AT 68

J. Irvine Lyle, a pioneer in the field of air conditioning, president of the Carrier Corp., Syracuse, N.Y., an active and ardent worker for the Society which he served as president in 1917 and in many other capacities from 1911 until his death, passed away at the age of 68 on June 7, 1942, at Syracuse, following an illness of 3 months.

Mr. Lyle was born in Fayette Co., Ky., February 14, 1874, and attended the public schools in Lexington. He received his B.M.E. in 1896 and his M.E. in 1902, from the University of Kentucky.

J. Irvine Lyle

He was first employed by the Pullman Co. at Ludlow, Ky., and then by the Southern Railway. Later he joined the Buffalo Forge Co. as a draftsman and in 1901 he became branch manager at New York, where he remained until 1908. It was during this period that Willis H. Carrier was assigned to New York as a sales engineer, and Mr. Lyle recognized his creative ability and sent Mr. Carrier back to Buffalo with the now famous remark, "I'm returning to you a fair-to-middlin' salesman in order that you may afford opportunity for the development of an unusually brilliant engineering genius." Mr. Lyle's prophecy is realized most strikingly in the fact that Mr. Carrier is, today, one of the outstanding engineers of the world.

In 1915, Mr. Lyle and Mr. Carrier, together with E. T. Murphy, M. S. Smith, L. L. Lewis, E. T. Lyle, E. P. Heckel and A. E. Stacey, Jr., realizing the possibilities and potentialities of scientific air conditioning, as distinguished from mere heating, cooling, or ventilating, founded the Carrier Engineering Corp., with its principal office in

New York City. As always, throughout his entire career Mr. Lyle chose to submerge his own name within that of the great engineer whose genius he so effectively fostered. With the executive capacity, the financial acumen, and the genius possessed by Mr. Lyle for selecting able men, the Carrier Engineering Corp. weathered the shoals of initial organization and finance to its present-day position. He served as treasurer and general manager of the company until 1930, when he conceived the merger of the Carrier Engineering Corp., York Heating and Ventilating Corp., and Brunswick-Kroeschell Co., aggregating assets of more than ten millions of dollars, and coordinating the operations of more than a dozen corporations in the air conditioning, refrigeration and unit heater industries. He became president and director of Carrier Corp. Since 1930 he also served as president, treasurer, and director of Aerofin Corp. and treasurer of Auditorium Conditioning Corp.

Mr. Lyle was a Past President of the *Refrigerating Machinery Association* and a member of the Executive Committee in 1935 and 1936.

Mr. Lyle was a leader not only in the business side of the air conditioning industry, but also in its technical progress. Numerous inventions, especially those dealing with atmospheric controls, are among his contributions to industry. He was the author of *Atmospheric Dehumidifying,* of *Humidity, Its Effect Upon Comfort and Health,* and many other technical papers bearing on air conditioning and related subjects will be found in the TRANSACTIONS of the ASHVE and in other technical publications.

He became a member of the ASHVE in 1911 and was elected a Life Member in January 1942. He served continually on numerous important committees from 1913 to 1930 and it was through his active and enthusiastic work that the Research Laboratory of the Society became a reality in 1919. In 1914 he was chairman of the Committee to Show Latest Developments and Progress in Heating and Ventilating Industrial Buildings, and for three years served on the Committee on Education and Publicity. In 1917 and 1918 he was chairman of the Committees to confer with both the U. S. Bureau of Mines and the *National District Heating Association,* and from 1919-1927 he served as chairman, honorary chairman, and vice-chairman of the Committee on Research, serving as vice-chairman of its Executive Committee in 1922 and as a member of this Committee in 1920, 1921 and 1923.

In 1919 he was a member of the Committee to Cooperate with the U. S. Navy Department. He also served as chairman of the Nominating Committee for two years, and on the Committee on Revision of Constitution for four years. He was a member of the Guide Publication Committee in 1922 and 1923 and gave much time and support to the successful launching of this major Society activity. He was always interested in the work of New York Chapter of which he was a member.

He served as a member of the Society's Council for two years before being elected President in 1917. Since that time he has been a member of the Advisory Council. He was also a member of the Advisory Committee for the First International Heating and Ventilating Exposition.

Mr. Lyle's interests extended beyond the engineering field and included the breeding of trotting horses, raising of pedigreed chickens, and the development of gardens. His famous Orpingtons were consistent winners in all the great American shows, in New York, Chicago, Philadelphia, Baltimore, Trenton, Boston, Augusta, Pittsburgh, and many other places. He served as president of the *New Jersey Poultry Association* and the *New Jersey Poultry Fanciers Association,* and in 1929, with Mrs. Lyle's assistance, Jill Farm won the New Jersey state garden award, a notable achievement in a state noted for its gardens and estates.

He was a director of the *Hambletonian Society* and treasurer of the Trotting Horse Club of America. Two of his horses were Brown Berry and Princess Peg, Hambletonian and Kentucky Futurity contestants.

Mr. Lyle was an engineer of great ability, a successful business man, a home lover, and a great friend. He preserved the genteel open-country vision of the old-time Kentuckian, coupled with the genial hospitality which has made his native state a synonym for graciousness. As one of his English guests remarked, "His home is your home and you only wish that you were a guest worthy of your host. Anyone can be successful, while only a handful can be, in addition, that altogether indescribable thing—a 'Kentucky gentleman.'"

From 1916 through 1924, Mr. Lyle served as a trustee of the University of Kentucky, and was a past president of its *Alumni Association.* He was a member of the Masonic Order, and of Sigma Chi and Tau Beta Pi fraternities. He was a member of the Cloud Club (New York), Century Club (Syracuse), Skaneateles Country Club, and the Congressional Country Club, Washington, D. C.

Mr. Lyle is survived by his widow, the former Elizabeth Biggerstaff of Lexington, Ky.; a daughter, Cornelia, now Mrs. Martin E. Snyder, Skaneateles, N. Y.; a son, Joel Irvine Lyle, III, Orlando, Fla.; and two brothers, C. R. Lyle, Washington, D. C., and E. T. Lyle, Orlando, Fla.

The Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS feel very keenly the loss of such an able and distinguished member and extend their sincerest and deepest sympathy to Mrs. Lyle and the family who survive.

Funeral services were conducted in Plainfield, N. J., on Tuesday afternoon, June 9, at 2:30 o'clock.

JOHN F. HALE, LIFE MEMBER AND PAST PRESIDENT, DIES

John F. Hale, a well-known figure in the heating and air conditioning field for the past 40 years, died on May 29, 1942, after an illness of several months' duration. He was born in St. Louis, Mo., November 11, 1870, and received his early education in the public schools of Chicago. He served his apprenticeship in Chicago with Burnham and Root, Theo. Jacobs Co., Charles H. Simmons Co. and Baker and Smith Co.

Mr. Hale was one of the pioneers in the development of vacuum heating, having created the engineering department of the Chicago office of Warren Webster & Co., with which he was associated from 1888 to 1902. During four of these years he acted as personal representative of Warren Webster, its president, with activities throughout the United States.

For a number of years he was Vice-President and General Manager of Consolidated Engineering Co., Chicago, Ill., and in 1917 he became President of Braemer Air Conditioning Corp., Philadelphia, Pa., which in 1919 became known as the Atmospheric Conditioning Corp. Until its acquisition by the Carrier Engineering Corp. in 1926, Mr. Hale was president. He joined the Aerofin Corp., Chicago, in 1926, serving as District Representative until 1929, at which time he became District Manager, the position he held until his death.

He joined the Society in 1902, served on the Board of Governors in 1908, 1910, 1912 and 1913. He was elected 1st Vice-President, and at that time he was requested to serve as acting president, taking the late John R. Allen's place, who was at that time absent from the United States in the development of the Engineering School, Roberts College, Constantinople. He was elected President of the Society in 1913 and had the unusual distinction of being in the president's chair for two years. Mr. Hale always maintained a lively interest in Society activities and was active in establishing the Illinois, New York and Philadelphia Chapters, having been a charter member of the Illinois Chapter of which he was Secretary during its first five years.

John F. Hale

During these early years in the Society, he served on numerous committees, as chairman of the Committee on Compulsory Legislation in 1909 and 1910; Committee on Engineering Cooperation, as chairman in 1917 and 1918; and in 1920, 1929 and 1930 he served as a member of the Nominating Committee.

In 1920 he also served as a member of the Board of Governors of the Eastern Pennsylvania Chapter, and from 1922-1925 as a member of the Committee on Revision of Constitution. In 1926 and 1927 he was a member of the Technical Advisory Committee on Temperature, Humidity and Air Motion, and on the Technical Advisory Committee on Effect of Wind and Weather Conditions on the Heating Loads in 1929. In 1935 he was a member of the Committee on Constitution and By-Laws.

The Officers and Council of the American Society of Heating and Ventilating Engineers extend their deepest sympathy to his widow and family for the loss they have sustained. He was one of the outstanding members of the Society and will be greatly missed by the many who knew him and by his business associates and colleagues in the Illinois Chapter of the Society.

ELWOOD WHITE DIES AT 55

Elwood Sanger White of Old Greenwich, Conn., president of the United States Radiator Corp., Detroit, Mich., died May 27, 1942 in Emergency Hospital, Washington, D. C., after a fortnight's illness.

Mr. White was born in Joliet, Ill., the son of the late Right Reverend John Hazen White, Episcopal Bishop of Northern Indiana, and Marie Holbrook White. He received his B.S. from Purdue University in 1909, and immediately became a salesman for the American Radiator Co., representing the company in Chicago, Boston and New York, until 1917, when he became general manager of the Edward R. Ladew Co., Glen Cove, L. I., a post he held until 1920.

The next year Mr. White became president of Taco Heaters, Inc., Stamford, Conn. In 1936 he was named president of the United States Radiator Corp. and in 1937 was elected president of the Pacific Steel Boiler Corp., Detroit.

Mr. White was a member of the Society for 22 years and he will be greatly missed by his many friends in the Society and his colleagues in the heating industry.

The Officers and Council of the American Society of Heating and Ventilating Engineers express their sincere sympathy to his widow, Mrs. Luella Perrin White, his daughter, Mrs. Hugh Rose, Jr., San Francisco, Calif., and to his son, John Hazen White, Old Greenwich.

Funeral services were held on May 29, at 2:30 o'clock, at St. John's Episcopal Church, Stamford, Conn., and the Rev. Stanley Hemsley officiated. Interment followed in Lafayette, Ind.

E. K. LANNING, WARREN WEBSTER ASSOCIATE, DIES

In the death of Emanuel K. Lanning, on May 26, 1942, at his home in Clayton, N. J., the heating industry lost one of its well-known figures. Mr. Lanning became associated with Warren Webster and Co., Camden, N. J., in 1903 as secretary to the late Theodore Webster, brother of the late Warren Webster, founder of the concern. Through the years he rose steadily in the organization and at the time of his death was sales manager, assistant secretary and a member of the Board of Directors.

Mr. Lanning was born in Philadelphia, Pa., October 2, 1882, receiving his early education in that vicinity. He was graduated from Girard College, Philadelphia, before beginning his business career.

In addition to Mr. Lanning's work with the Webster Co., he was active in the national affairs of the steam heating specialties industry. For some years he served as president of the *Steam Heating Equipment Manufacturers Association.* During the NRA days, he was selected as chairman of the Code Authority for the steam heating equipment industry. He joined the Society in 1927 and was active in the Philadelphia Chapter.

At the time of his death, Mr. Lanning was a member of the School Board of Clayton, N. J., where he lived for over 20 years, and was a member of the Forest Lodge of Masons, Trinity Methodist Church and served on the official board of the Church.

Mr. Lanning is survived by his widow, Mrs. Mignon Mae Lanning, a sister and two grandchildren, to whom the Officers and Council of the American Society of Heating and Ventilating Engineers extend their heartfelt sympathy in their bereavement.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 14 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by July 15, 1942, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES *Proposers*	 *Seconders*
BEERS, LOUIS N., Heating Engr., W. T. Andrew Co., Detroit, Mich.	L. A. Burch M. B. Shea	R. L. Deppmann F. X. Marzolf
BORNSTEIN, ALFRED B., Partner, William Bornstein & Son, Washington, D. C.	M. S. Hall Vernon Graves	F. M. Thuney William Bornstein
BURNS, FRED C., Manager, Kewanee Boiler Corporation, Kansas City, Mo.	P. C. Leffel Gustav Nottberg	K. M. Stevens W. A. Russell
CLEMENT, EUGENE R., SR., Sales & Engrg., E. R. Clement, Inc., Bridgeport, Conn. (*Reinstatement*)	R. W. Cumming E. J. Ritchie	L. A. Teasdale C. J. Lyons
CLOW, SHERWOOD A., Sales Engr., Buffalo Forge Co., Dallas, Tex.	T. H. Anspacher V. G. Cox	L. C. McClanahan L. S. Gilbert
COHN, HENRY, Mechanical Engr., Giffels & Vallet, Detroit, Mich.	E. F. Hyde R. S. M. Wilde	B. A. King, Jr. D. E. Siegel
GILLILAND, LESLEY L., Operating Engr., Gadsden Ordnance Plant, Gadsden, Ala.	H. C. Gause J. A. Cunninghame (*Non-member*)	M. R. Thomas (*Non-member*) S. C. Bratton (*Non-member*)
GUNZEL, RUDOLPH M., Rep., R. M. Gunzel & Co., Los Angeles, Calif. (*Reinstatement*)	Warren Webster, Jr. Kessler Webster	R. B. Dickson F. G. Cross
HART, JOHN H., Mechanical Engr., Federal Housing Administration, Washington, D. C.	R. S. Dill R. K. Thulman	J. H. Fogg F. M. Thuney
KIRKENDALL, HORTON J., Mfrs. Sales Repr., Pittsburgh, Pa. (*Advancement*)	A. H. Schroth J. F. Collins, Jr.	P. A. Edwards A. A. Marks
NATION, OSLO, Associate Engr., U. S. Engineering Defense Div., Denison, Tex.	W. A. Russell G. A. Linskie	J. V. Edmonson (*Non-member*) E. N. McDonnell
OTT, M. EARL, Pres., Parkside Sheet Metal Works, Inc., Chicago, Ill.	S. L. Haas J. S. Kearney	E. M. Mittendorff A. O. May
RILEY, JOHN N., Mgr., Danforth Co., Pittsburgh, Pa.	G. G. Waters E. C. Smyers	R. H. Sweeney T. F. Rockwell
SCHAUER, ROBERT A., Htg. & Air Cond. Engr., Westinghouse Electric & Manufacturing Co., Canton, O.	T. F. Rockwell F. C. McIntosh	Paul Gayman L. E. Slawson

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

BARTELS, CHARLES J., Owner, Automatic Stoker & Engineering Co., Parkersburg, W. Va.

BAXTER, JULIAN F., Pres., Automatic Coal Burning Corp., Atlanta, Ga.

BEVINGTON, WARREN C., Pres., Bevington-Williams, Inc., Indianapolis, Ind. (*Reinstatement*)

BURKE, JOHN S., Dealer Coordinator, New Orleans Public Service Inc., New Orleans, La.

GORDON, WILLIAM D., Vice-Pres.-Gen. Mgr., Canada Register & Grille Co., Ltd., Toronto, Ont., Canada (*Reinstatement*)

HOLE, WILLIAM G., Mgr., (Air Filter Div.) Darling Bros., Ltd., Montreal, Que., Canada

KAMMAN, ARNOLD R., Owner, Arnold R. Kamman Co., Buffalo, N. Y. (*Advancement*)

LANDFRIED, CHARLES L., Ventilating Engr., Gielow, Inc., New York, N. Y.

LANING, BENJAMIN A., Utilities Officer, Veterans' Administration, Washington, D. C.

LEHMAN, JOHN L., Sales Engr., Columbia Radiator Co., East Orange, N. J.

McGOWAN, THOMAS E., Mech. Engr., Basic Magnesium Inc., Boulder City, Nev.

MORRIS, EDWARD J., Manager, Morris Engineering Co., Baltimore, Md.

MURPHY, CHARLES G., Ventilating Engr., Wright Aero, Lockland, Ohio (*Reinstatement*)

PINNEY, THEODORE M., General Projects Engr., The Austin Co., Cleveland, O.

RAY, JOHN A., Sales Engr., W. E. Lewis & Co., Dallas, Tex.

Rink, Charles N., Sales Mgr. (Coil Div.), McQuay Inc., Minneapolis, Minn.

Shelldrop, Tonn F., Squad Boss (Mech. Dept.), Caribbean Architect, New York, N. Y.

Smith, Roger K., Instructor, Iowa State College, Ames, Ia.

Thompson, John, Building Engr., Hydro-Electric Power Commission, Toronto, Ont., Canada.

Todd, Malcolm McM., Owner, Malcolm Todd, Toronto, Ont., Canada

Turner, John P., Jr., Dist. Repr., General Electric Co., Portland, Ore.

ASSOCIATES

Bazzoni, Joseph P., Mech. Engr., G. P. Rice Co., Biloxi, Miss.

Cummings, Thomas P., Sales Engr., Hayes Furnace Manufacturing Co., Los Angeles, Calif.

deMena L., Isabel, Student, National School, Los Angeles, Calif.

DeVries, John J., Insulation, Grand Rapids, Mich.

McWilliams, Joseph W., Mechanical Engr., Eastman Kodak Co., Rochester, N. Y.

Padden, Bernard, Research Engr., Anemostat Corporation of America, New York, N. Y.

Schumacher, Clarence W., Design Engr., Battey & Childs & Consoer-Townsend & Quinlan, Parsons, Kan.

Segle, Thomas L., Sales Engr., Wells & Wade, Inc., Wenatchee, Wash.

Tichenor, Leslie R., Jr., Vice-Pres.-Treas., L. R. Tichenor & Son, Hillside, N. J. (*Reinstatement*)

Tierney, Lawrence J. J., Owner, L. J. Tierney Co., Boston, Mass. (*Reinstatement*)

JUNIORS

DeFlon, James G., Cooling Tower Engr., The Fluor Corp., Los Angeles, Calif.

Dominy, Charles B., Junior Engr., Puget Sound Navy Yard, Bremerton, Wash.

Fly, Elmer P., Shop Supt., George M. Fly & Son, Inc., Nashville, Tenn.

Gould, James L., Erection Supt., Cooke-Bain Air Conditioning & Sheet Metal Contractors., Tulsa, Okla.

Officers, Council and Committees

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
51 Madison Ave., New York, N. Y.

OFFICERS

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First Vice-President.......................................M. F. BLANKIN
Second Vice-President.......................................S. H. DOWNS
Treasurer.......................................E. K. CAMPBELL
Secretary.......................................A. V. HUTCHINSON
Technical Secretary.......................................JOHN JAMES

COUNCIL

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Three Years: L. G. MILLER, A. J. OFFNER, A. E. STACEY, JR., B. M. WOODS.

Two Years: A. P. KRATZ, W. A. RUSSELL, L. P. SAUNDERS, C. TASKER.

One Year: J. F. COLLINS, JR., W. L. FLEISHER, E. N. MCDONNELL, T. H. URDAHL, C.-E. A. WINSLOW, F. C. MCINTOSH, *Ex-Officio*.

COUNCIL COMMITTEES

Executive—W. L. Fleisher, *Chairman*, A. J. Offner, A. E. Stacey Jr.

Finance—E. N. McDonnell, *Chairman*, J. F. Collins, Jr., C.-E. A. Winslow.

Membership—W. A. Russell, *Chairman*, M. F. Blankin, B. M. Woods.

Standards—S. H. Downs, *Chairman*, L. G. Miller, C. Tasker.

Meetings—A. P. Kratz, *Chairman*, L. P. Saunders, T. H. Urdahl.

ADVISORY COUNCIL

W. L. Fleisher, *Chairman;* Homer Addams, D. S. Boyden, W. H. Carrier, S. E. Dibble, W. H. Driscoll, F. E. Giesecke, H. P. Gant, E. Holt Gurney, L. A. Harding, H. M. Hart, C. V. Haynes, E. Vernon Hill, John Howatt, W. T. Jones, D. D. Kimball, G. L. Larson, S. R. Lewis, Thornton Lewis, J. F. McIntire, F. B. Rowley and A. C. Willard.

SPECIAL COMMITTEES

Admission and Advancement—Earle W. Gray, *Chairman* (*one year*); E. P. Heckel (*two years*), T. T. Tucker (*three years*).

ASHVE-IES Joint Committee on Lighting and Air Conditioning—H. M. Sharp, *Chairman*; W. R. Beach, B. C. Candee, W. G. Darley, C. L. Kribs, Jr., P. M. Rutherford, Jr.

Chapter Relations—J. F. Collins, Jr., *Chairman;* Lincoln Bouillon, M. M. Rivard.

Constitution and By-Laws—Albert Buenger, *Chairman*, L. T. Avery, R. H. Carpenter.

Committee on Code for Testing Surface Coils for Heating and Cooling—C. A. McKeeman, *Chairman;* Tom Brown, E. L. Hogan, H. F. Hutzel, L. E. Moody, A. F. Nass, S. F. Nicoll, M. Noble, L. P. Saunders, and D. C. Wiley.

Committee on Safety Regulations for Heating, Ventilating and Air Conditioning Systems—N. A. Hollister, *Chairman;* F. H. Buzzard, B. F. McLouth, G. P. Nachman, and C. H. Randolph.

Committee on ASHVE Code for Testing Air Cleaning Devices—John Howatt, *Chairman;* O. W. Armspach, J. L. Blackshaw, R. S. Dill, S. P. Eagleton, S. R. Lewis, Arthur Nutting, G. W. Penney, F. B. Rowley, J. H. Waggoner, and Dr. Leonard Greenburg, *Ex-Officio*.

Guide Publication—A. J. Offner, *Chairman*, P. D. Close, S. Konzo, C. S. Leopold, G. L. Tuve.

F. Paul Anderson Award—M. F. Blankin, *Chairman*, W. H. Carrier, J. D. Kroeker, J. F. McIntire, F. B. Rowley.

Joint Committee on Standards for Residence Heating—L. E. Seeley, *Chairman;* R. S. Dill, S. H. Downs, H. L. Dryden, J. G. Eadie, H. C. Meyer, Jr., R. K. Thulman, T. H. Urdahl, and H. Vermillyea.

Joint Committee to Study Uniform Methods of Heat Loss Calculation—P. D. Close, *Chairman;* A. P. Kratz, J. P. Magos, G. P. Nachman, A. F. Nass.

Joint Committee on Weather Statistics—J. F. Collins, Jr., *NDHA, Chairman;* F. H. Dewey, *OBI*, C. M. Humphreys, ASHVE, S. Konzo, *NWAHACA*, Raymond Little, *AGA*, Dr. E. Dillen Smith, *U. S. Weather Bureau*.

Publication—James Holt, *Chairman* (*one year*); C. H. B. Hotchkiss (*two years*), J. H. Walker (*three years*).

War Service Committee—B. M. Woods, *Chairman*, W. A. Danielson, W. L. Fleisher, E. Holt Gurney, Thornton Lewis, A. C. Willard.

NOMINATING COMMITTEE 1942

Chapter	*Representative*	*Alternate*
Atlanta	T. T. Tucker	A. H. Koch
Cincinnati	C. E. Hust	Albert Buenger
Connecticut	L. E. Seeley	Stanley Hart
Delta	L. V. Cressy	J. S. Adair
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